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## THE PITOT TUBE APPLIED TO THE MEASUREMENT OF AIR.


BY LEO LOEB, M. E., ASSOCIATE.


There was recently submitted for test at the Engineering Experiment Station, Annapolis, Md., a new type of air heater proposed for installation on shipboard. This heater consisted of a number of tubes constructed on the film principle, wherein the air to be heated was passed upward inside of the inner tubes and outside of the outer tubes, and the steam supplying the heat passed down the film between the tubes. The heating surface was supposed to be rendered more efficient by the addition of retarders. Each retarder consisted of a helix of sheet copper brazed to the tube.

The heater was to be tested under a wide range of conditions in which both the pressure of the heating steam and the velocity of the heated air were varied, and it became necessary to find a simple and accurate method of metering the quantity of air supplied the heater in order to obtain a measure of the sensible heat transmitted through the tube surface. The selection of Pitot tubes for this purpose was based purely

on facility and convenience in arrangement of test apparatus, as the Venturi and electric meters are equally accurate for this work.

Thereupon there was undertaken a study of the use of the Pitot tube under various conditions; and the work herein presented, which is largely collected from other sources, forms a rather complete survey of the use of the Pitot tube in the measurement of air, and is given with the hope that it will enable others who have the same problem to face to obtain all the desired information thus collected without recourse to the numerous original sources.

The final point in the research was the practicability of substituting a single Pitot tube at the center of a circular duct for a number of tubes distributed over the cross-section, or for surveys on various diameters; and as there were no data available on this point, a series of tests were undertaken to determine the ratio of average velocity to center velocity at velocities between 1,000 and 3,000 feet per minute. This portion of the work, as far as the writer's knowledge of the literature on the subject is concerned, is a new contribution; and it is hoped in time to continue the investigations to other forms and proportions of ducts.

The Pitot tube was invented by the French geometer and engineer, Henri Pitot, from whom it derives its name, who used it as early as 1732 for measuring the velocity of water in open channels. The use of the tube was made more general by Darcy who, in 1856, provided the additional tube for measuring the static pressure.

In its simplest form the Pitot tube, as shown in Fig. 1, consists of a tube bent at right angles with the open end A turned in such a manner as to be opposed to the direction of flow of the fluid in the channel. A manometer attached to the end C measures in suitable units the pressure in the tube. If the substance which is flowing at uniform velocity in the channel B is noncompressible and is under no pressure other than that of the atmosphere which surrounds it, then its velocity of flow is given by the relation

$$V = \sqrt{2gh},$$

where

$V$ is velocity in feet per second,

$g$ is the acceleration due to gravity = 32.16 feet per second, per second,

$h$ is the head indicated by the manometer when reduced to equivalent feet of the substance measured.

If the manometer registers in feet of water the velocity will be given by the relation

$$V = \sqrt{\frac{2gh}{s}},$$

where

$h$ is the head in feet of water, and

$s$ the specific weight referred to water as unity; $\frac{h}{s}$ is the number of feet of the fluid which would give the same pressure as $h$ feet of water.

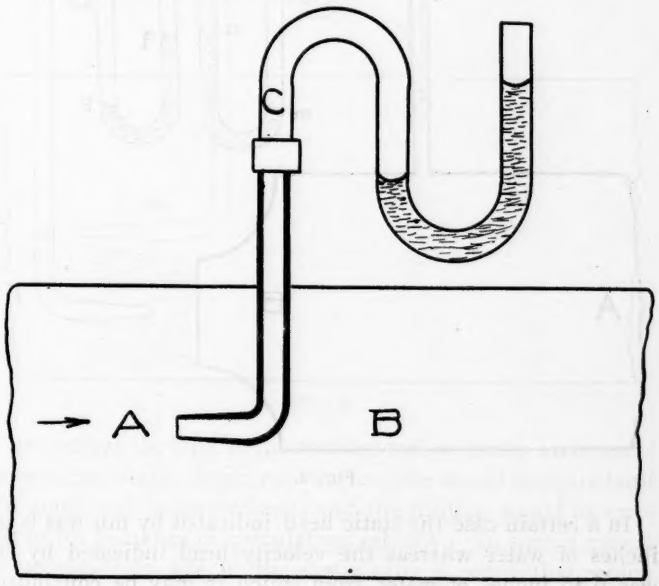



FIG. 1.

The correctness of this theory may be verified by a simple experiment indicated in the accompanying Fig. 2. AB is a duct with a converging nozzle from which air is flowing at a uniform rate. The static pressure of the air in the duct at a point where the velocity is negligible compared to the nozzle velocity is indicated by the elevation mn of a column of water in the U-tube, C. The Pitot tube, T, is mounted to show the velocity of the issuing jet by means of the water manometer at D. If the Pitot tube bears out in practice its theory as given above, then it may be said to have a coefficient of unity. As a matter of fact the column pq will be slightly less than the static head mn, due to the friction loss in the nozzle and not to any defect in the Pitot tube, for since the tube merely transmits pressure to the manometer without a flow of fluid, there can be no attendant friction loss.

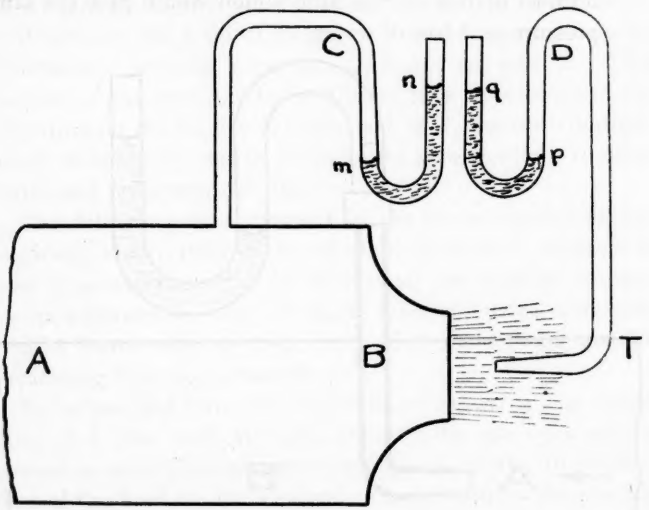



FIG. 2.

In a certain case the static head indicated by mn was 6.30 inches of water whereas the velocity head indicated by pq was 6.26 inches of water, from which it may be concluded that:

1. The Pitot tube placed in a stream discharging into the free atmosphere indicates a total pressure equal to the static pressure plus the velocity head $\left(h = \frac{V^2}{2g}\right)$ at the point where the static pressure is taken;

2. The loss of energy from friction in the nozzle in the case of air is very slight, somewhat less than 1 part in 100, and consequently the velocity loss does not exceed the half of this, or 0.5 per cent.

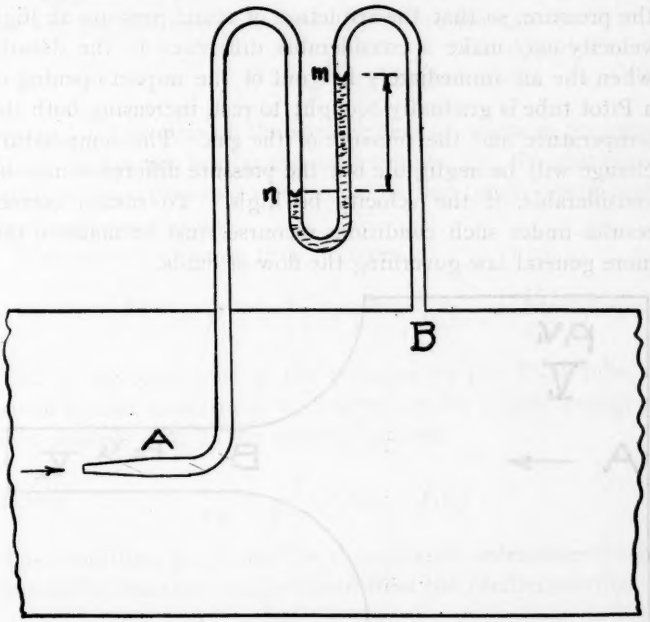



FIG. 3.

In general the fluid in the channel will be under some static pressure so that a single manometer tube would measure both the static and dynamic heads and the reading would be erroneous for purposes of calculating velocity. In such a case it is necessary to add to the static tube a connection which indicates or balances the static pressure. A in Fig. 3 is the dynamic connection and B the static connection. It follows

that the differential head mn indicated on the U tube is a measure of the velocity head provided proper precautions have been taken to eliminate any aspirating effect past the open end, B. The methods of guarding against this will be described on pages 1133-34, under the section devoted to types of tubes.

Thus far all relations have been deduced on the assumption that the fluid was noncompressible. However, air is the typical perfect gas whose density at constant temperature varies as the pressure, so that the reduction of static pressure at high velocity may make a considerable difference in the density when the air immediately in front of the impact opening of a Pitot tube is gradually brought to rest, increasing both the temperature and the pressure of the gas. The temperature change will be negligible but the pressure difference may be considerable, if the velocity be high. To obtain correct results under such conditions recourse must be made to the more general law governing the flow of fluids.

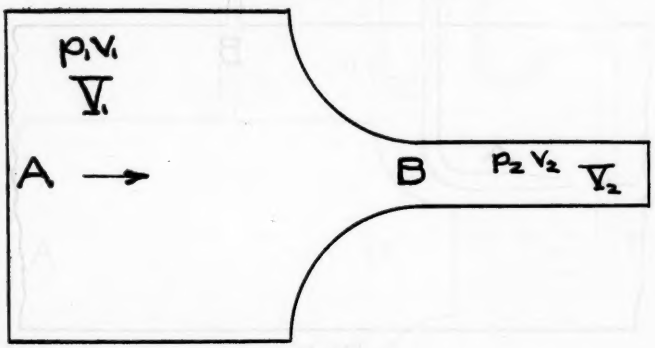



FIG. 4.

Suppose a gas to be flowing through a pipe, A (Fig. 4), of large diameter and to enter by gradual change a pipe, B, of much smaller diameter. The gas in A has a velocity $V_1$, a pressure of $p_1$ and a specific volume of $v_1$ cubic feet per pound. For each unit of weight, say one pound, that passes any point in A an equal weight passes a section in B with a velocity $V_2$,

a pressure $p_2$ and a specific volume of $v_2$ cubic feet per pound. The same amount of energy exists in a unit weight on one side as on the other, so we may write the energy equation:

$$E_1 + p_1v_1 + \frac{V_1^2}{2g} = E_2 + p_2v_2 + \frac{V_2^2}{2g}, \quad . \quad . \quad . \quad (1)$$

in which the terms $E_1$ and $E_2$ represent the internal or "intrinsic" energy of the substance, $p_1v_1$, the pressure energy, and $\frac{V_1^2}{2g}$ the kinetic energy of one pound of gas. For a perfect gas the internal energy is given by the relation:

$$E = \frac{pv}{k-1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $k$ is the ratio of the specific heat at constant pressure, $C_p$, to the specific heat at constant volume, $C_v$; the numerical value of the ratio being 1.405 under normal conditions of pressure and temperature.

The energy equation then becomes:

$$\frac{p_1v_1}{k-1} + p_1v_1 + \frac{V_1^2}{2g} = \frac{p_2v_2}{k-1} + p_2v_2 + \frac{V_2^2}{2g}. \quad . \quad . \quad (3)$$

Taking the conditions at the entrance to the Pitot tube, as given by the subscript 2, the expression for kinetic energy at that point is zero, as the velocity is zero.

Hence,
$$\frac{V_1^2}{2g} = \frac{k}{k-1}(p_2v_2 - p_1v_1). \quad . \quad . \quad . \quad . \quad (4)$$

The conditions $p_2v_2$ cannot be conveniently determined experimentally, but they can be found from the further relation

$$p_2v_2^k = p_1v_1^k, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

the equation for adiabatic expansion of a gas. Then

$$p_2v_2 = p_1v_1\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and

$$\frac{V_1^2}{2g} = p_1v_1\frac{k}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k}{k-1}} - 1\right]. \quad . \quad . \quad . \quad . \quad (7)$$

The characteristic equation of a perfect gas, $pv = RT$, in which $R$ is a constant, for air 53.35, and $T$ the absolute temperature, 459.5 plus the Fahrenheit temperature, is used to further simplify equation 7, as follows:

$$\frac{V^2}{2g} = RT\frac{k}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right]. \quad . \quad . \quad . \quad . \quad (8)$$

Substituting numerical values:

$$\begin{aligned} g &= 32.16, \\ R &= 53.35, \\ k &= 1.405, \\ t &= 70 \text{ degrees F.}, \\ T &= 529.5 \text{ degrees F.}, \end{aligned}$$

$$V^2 = 11{,}904\ T\left[\left(\frac{p_2}{p_1}\right)^{.288} - 1\right], \quad . \quad . \quad . \quad . \quad (9)$$

or

$$V = 2{,}511.6\sqrt{\left[\left(\frac{p_2}{p_1}\right)^{.288} - 1\right]}, \quad . \quad . \quad . \quad (10)$$

$V$ being in feet per second.

Since the pressures in this equation enter as a ratio, the value of the velocity is independent of the system of units in which the pressures are measured, provided both $p_2$ and $p_1$ are indicated as absolute pressures. Thus, if both the static and velocity heads are measured in inches of mercury and the barometer reading at the time is 30 inches of mercury, then the ratio $\frac{p_2}{p_1}$ would be expressed as

$$\frac{30 + h_2}{30 + h_1},$$

where $h_2$ is the static head and

$h_1$ is the static head plus the dynamic head.

The reduction of the simple velocity formula $V = \sqrt{2gh}$ to a working form when $h$ is measured in inches of mercury is made with the use of the following constants:

Weight of a cubic foot of dry air at atmospheric pressure and 70 degrees F. temperature = .07495;

Density of mercury = 13.59;

Weight of a cubic foot of water = 62.38 pounds;

$$g = 32.16.$$

Hence

$$V = \sqrt{2 \times 32.16 \times \frac{13.59}{12} \times \frac{62.38}{.07495} h};$$

$$= 246.22 \sqrt{h}. \quad \ldots \ldots \ldots \ldots \quad (11)$$

As a comparison of the two formulae 10 and 11, the velocities corresponding to a static head of 30.00 inches of mercury absolute and velocity heads of 0.2, 0.3, 1.8 and 3.2 inches of mercury have been computed and are shown in Table I, while Fig. 5 shows the velocities by approximate and by exact formulae plotted as a function of dynamic head.

TABLE I.

| Velocity head. Inches of Hg. | Velocity in feet per second. | | Velocity in feet per minute. | | Per ct. difference. |
|---|---|---|---|---|---|
| | Formula 10. | Formula 11. | Formula 10. | Formula 11. | |
| 0.2 | 109.87 | 110.25 | 6,592 | 6,615 | 0.35 |
| 0.8 | 219.30 | 220.50 | 13,158 | 13,230 | 0.55 |
| 1.8 | 326.58 | 330.75 | 19,595 | 19,845 | 1.28 |
| 3.2 | 432.09 | 441.00 | 25,925 | 26,460 | 2.06 |

It is to be noted that for velocities below 10,000 feet per minute there is no appreciable difference between the two curves, and there is consequently no gain in accuracy commensurate with the laborious calculation from formula 10. In fact there may be a sacrifice, for the exact method requires the use of a six or seven-place table of logarithms to maintain the required precision in the term $\sqrt{\left[\left(\frac{p_2}{p_1}\right)^{.288} - 1\right]}$.

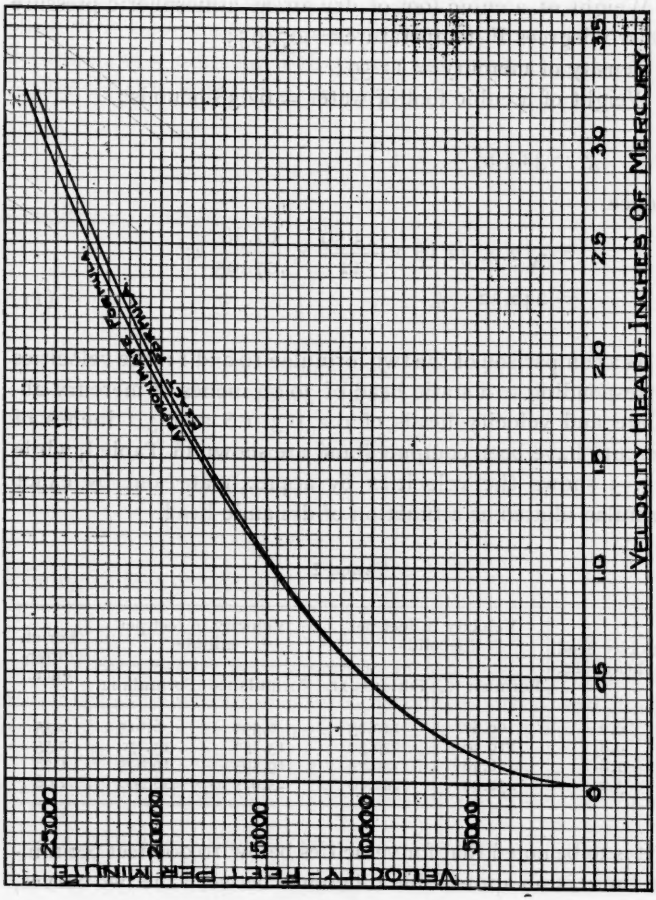



FIG. 5.—RELATION BETWEEN APPROXIMATE AND EXACT FORMULAE.

There seems to be one possible objection to the derivation of the formula discussed above, and that is that instead of the air being reduced to a state of rest at the entrance to the tube ($V_2 = 0$), the stream lines are simply divided and the air continues on its course at the same velocity after being deflected by the obstructing tube. In that case the effect is simply one of impact, and the formula which has been called "approxi-

mate" becomes "exact," since the differential gage will measure the velocity head of the air under the temperature and pressure conditions found in the duct. The close agreement of two methods of calculating flow, one based on dynamic, the other on thermodynamic reasoning, would tend to substantiate the theory.

## EFFECT OF MOISTURE ON AIR DENSITY.

Thus far the discussion has assumed the air to be dry. Conditions of air entirely free from moisture are never found, and in certain cases the change in density of the air due to large percentages of water vapor may make a considerable difference in velocity calculations. It is not the province of this article to go into the details of hygrometry, but it should be noted that more care is required in the determination of relative humidity than has usually been taken in the measurement of that quantity.*

The ordinary wet and dry-bulb thermometer and the tables usually used with it are far from accurate on account of the great influence of the velocity of currents of air in which the apparatus may be exposed as well as radiation from surrounding objects. Precise results can be attained only by the use of a good form of sling or aspiration psychrometer or a dew-point apparatus such as the Alluard.

According to Dalton's law, the pressure sustained by a mixture of two vapors or gases is the sum of the pressures sustained by each when occupying the volume of the mixture. From this the density of the moist air in pounds per cubic foot may be at once determined when the relative humidity, temperature and absolute pressure are known.

Let

$t$ = the temperature of the moist air ;

$p$ = absolute static pressure of the air, measured in pounds per square inch ;

$X$ = relative humidity, expressed in hundredths ;

* Rational Psychrometric Formula, Willis H. Carrier, "Journal A. S. M. E.," November, 1911, page 1309.

$p'$ = pressure of saturated vapor corresponding to temperature $t$, as given by a standard steam table;

$s$ = density, in pounds per cubic foot of dry saturated vapor at temperature $t$.

Then the pressure sustained by the vapor in the mixture of air and vapor is $Xp'$ pounds per square inch, and that sustained by dry air is

$$p - Xp' \text{ pounds per square inch.}$$

The density of the vapor is

$$Xs \text{ pounds per cubic foot,}$$

and the weight in pounds of a cubic foot of air free of moisture is

$$m = \frac{(p - Xp') \times 144 \times 1}{53.35\,(459.5 + t)};$$

and the density of the moist air is

$$Xs + \frac{(p - Xp') \times 144}{53.35\,(459.5 + t)}.$$

As an application of the above method assume the following conditions:

| | |
|---|---|
| Corrected barometer, inches of mercury, . . . | 29.98 |
| Static air pressure, inches of mercury, . . . . | 0.92 |
| Temperature of air, degrees F., . . . . . | 75.5 |
| Relative humidity, . . . . . | 0.56 |

The absolute air pressure is then

$(29.98 + 0.92) = 30.90$ inches of mercury, or 15.18 pounds per square inch.

From steam tables it is found that the vapor pressure corresponding to 75.5 degrees is 0.43 pound per square inch, and the vapor density is .00137 pound per cubic foot. Then the pressure sustained by the moisture in this mixture is

$.56 \times .43 = .24$ pound per square inch, and the weight of vapor associated with one cubic foot of moist air is

$.56 \times .00137 = .00077$ pound.

Then the dry air has a pressure of
15.18 — .24 = 14.94 pounds per square inch, and the weight of dry air associated with one cubic foot of moist air is

$$m = \frac{14.94 \times 144}{53.35 \times 535} = .07537 \text{ pound.}$$

The true density of the air is .07537 + .00077 = .07610 pound per cubic foot.

The effect of varying proportions of moisture on the density of humid air is shown by the following table, which assumes total absolute pressure of 30 inches of mercury and a temperature of 70 degrees F.:

TABLE II.

| Relative humidity. | Vapor pressure, lbs. per square inch. | Air pressure. | Weight of water vapor. | Weight of dry air. | Density of moist air, lbs. per cubic foot. |
|---|---|---|---|---|---|
| 0 | 0.00 | 14.74 | .00000 | .07514 | .07513 |
| 20 | 0.07 | 14.67 | .00023 | .07478 | .07501 |
| 40 | 0.14 | 14.60 | .00046 | .07442 | .07486 |
| 60 | 0.22 | 14.52 | .00069 | .07401 | .07470 |
| 80 | 0.29 | 14.45 | .00092 | .07366 | .07458 |
| 100 | 0.36 | 14.38 | .00115 | .07330 | .07445 |

The maximum variation for this case is 0.9 per cent.

This effect is also shown graphically in Fig. 6 (p. 1128).

There seldom arises in practice a case where the velocity of the stream of air is constant across the cross section through which it flows; the situation is further complicated by the fact that the velocity at any one point in a cross section does not remain the same, due to fluctuations in either demand or performance of the fan supply, so that some attention should be directed to the effect of such fluctuations on Pitot tube measurements. A very complete investigation of one phase of the problem relation between a series of instantaneous observations and a mean velocity has been made by Prof. M. Rateau and reported in "Annales des Mines," March, 1898. The plan of his experiments was to use a Pitot tube in two parallel currents of different velocities and compare individual readings with observations taken when the tube was rapidly

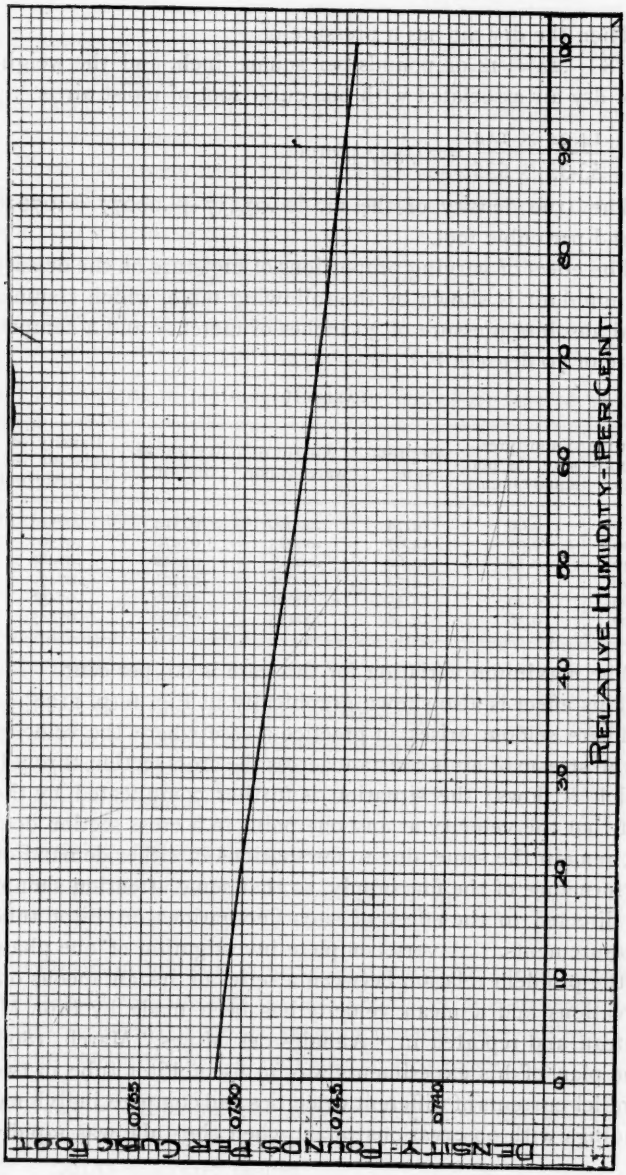



FIG. 6.—VARIATION IN DENSITY OF AIR WITH RELATIVE HUMIDITY.

moved from one stream to the other. For this purpose he devised the arrangement shown in Fig. 7. A casing AB is divided by a longitudinal partition MN into two compartments in which different velocites are maintained by suitable arrangement of dampers and fans. Air escapes from the nozzle common to both compartments in such a way as to give two adjacent streams a and b, in which for at least a short distance from the orifice, xy, the velocity in each of them is quite uniform, and in which the static pressure is always equal to the pressure of the surrounding atmosphere. Then a single-impact tube could be used to measure the velocity in the streams. This tube was moved back and forth across the outlet area by means of a crank and connecting-rod gear which could be turned at any desired speed. Thus the effect was the same as if the tube had received al-

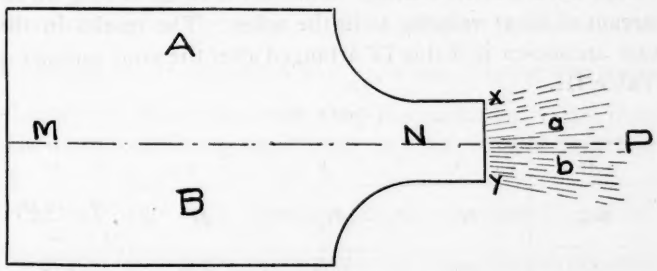



FIG. 7.

ternately and for equal times impulses from two currents of air possessing the velocities $V_1$ and $V_2$. For alternations below two per second the oscillation of the manometer column was visible, but above that a steady reading to half millimeters (.02 inch) could be obtained by throttling the passage to the manometer column. The results of Prof. Rateau's experiments are given in Table III (p. 1130), in which

$H_1$ is the velocity head in inches of water in jet a;

$H_2$ is the velocity head in inches of water in jet b;

$H_3$, column 4, is the velocity head obtained by immersing the tube for equal intervals, first in a then in b, and column 5 gives the average of $H_1$ and $H_2$.

TABLE III.

| 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|
| No. | $H_1$ | $H_2$ | $H_3$ | $\frac{H_1+H_2}{2}$ |
| 1 | 3.46 | 2.24 | 2.84 | 2.85 |
| 2 | 3.19 | 1.38 | 2.28 | 2.28 |
| 3 | 3.48 | 1.54 | 2.52 | 2.51 |
| 4 | 3.34 | .98 | 2.16 | 2.16 |
| 5 | 3.25 | .63 | 1.93 | 1.94 |
| 6 | 3.25 | .18 | 1.69 | 1.71 |

A comparison of columns 4 and 5 shows that the Pitot tube indicates for this case the arithmetic mean between the dynamic heads due to the velocities $V_1$ and $V_2$.

Prof. Rateau then modified his apparatus so that the tube would have a motion which immersed it twice as long in the stream of lesser velocity as in the other. The results in this case are shown in Table IV arranged after the same manner as Table III.

TABLE IV.

| 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|
| No. | $H_1$ | $H_2$ | $H_3$ | $\frac{H_1+2H_2}{3}$ |
| 7 | 3.03 | 0.16 | 1.06 | 1.12 |
| 8 | 3.31 | 0.85 | 1.69 | 1.67 |
| 9 | 3.39 | 1.18 | 1.89 | 1.92 |
| 10 | 3.46 | 1.22 | 1.97 | 1.97 |
| 11 | 3.86 | 2.16 | 2.72 | 2.73 |

Except for the value in experiment 7 the general conclusion to be drawn from these data is that the observed dynamic pressure coincides with the mean of the pressures due to $V_1$ and $V_2$, taking into account the difference between the time the tube is subjected to these velocities. Considering only the results given in Table III it is quite evident from the deduced data in Table V that the Pitot tube is liable to show very exaggerated velocities, especially when applied to irregular flows. The error in the velocity (column 7) is shown as

TABLE V.

| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|
| No. | $V_1$ | $V_2$ | $V$ | $\frac{V_1 + V_2}{2}$ | $\frac{V_2}{V_1}$ | Error, per cent. | $a$ |
| 1 | 7,550 | 6,060 | 6,830 | 6,790 | 0.81 | 0.6 | 1.012 |
| 2 | 7,230 | 4,745 | 6,120 | 5,985 | 0.66 | 2.3 | 1.046 |
| 3 | 7,760 | 5,020 | 6,415 | 6,280 | 0.67 | 2.2 | 1.044 |
| 4 | 7,405 | 4,010 | 5,945 | 5,710 | 0.54 | 4.1 | 1.082 |
| 5 | 7,310 | 3,208 | 5,630 | 5,255 | 0.44 | 7.1 | 1.143 |
| 6 | 7,310 | 1,693 | 5,280 | 4,510 | 0.23 | 17.1 | 1.370 |

a function of the ratio between $V_2$ and $V_1$ (column 6); $V$, in column 4, is the apparent velocity, in feet per minute, indicated by the Pitot tube oscillating between the two streams, whereas $\frac{V_1 + V_2}{2}$ (column 5) is the true mean velocity. The last column gives the value of the coefficient $a$, which must be inserted in the formula $h = a\frac{V^2}{2g}$, where $V$ is the true mean velocity. It should be noted that the coefficient varies over wide limits, increasing rapidly for velocity ratios of less than 0.5.

Figure 8 (page 1132) shows graphically the preceding table. The ratio $\frac{V_2}{V_1}$ is plotted as abscissae and the ordinates give the relative error in per cent. shown by the Pitot tube.

So this instrument, which is extremely accurate in the case of a uniform current, is only approximately correct when applied to the measurement of an average of variable velocities. As long as the durations of the velocities $V_2$ and $V_1$ are equal, the indications correspond not to the average mean velocity but to the average of the squares of the velocities $V_1$ and $V_2$, and probably this relation extends to the more general case where the velocities vary in any manner; that is to say the Pitot tube is an instrument which measures the energy and not the velocity of a stream.

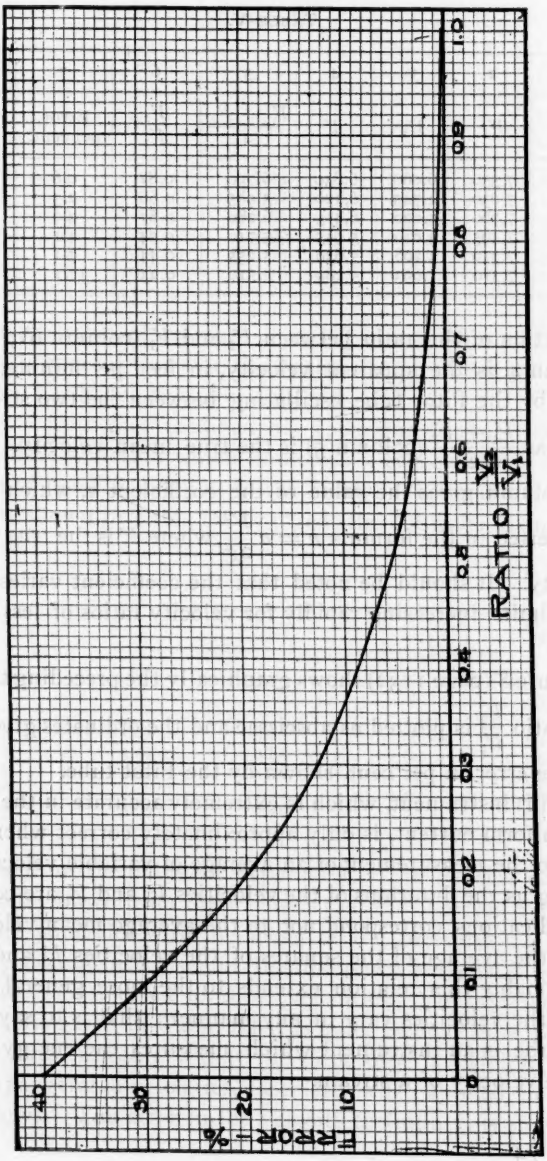



FIG. 8.—ERROR IN MEAN VELOCITY DUE TO VARIABLE VELOCITIES.

TYPES OF TUBES AND PRESSURE-INDICATING DEVICES.

One of the most successful forms of tubes for moderate velocities is the tube developed by Naval Constructor D. W. Taylor, U. S. Navy. This tube, shown in Fig. 9, combines

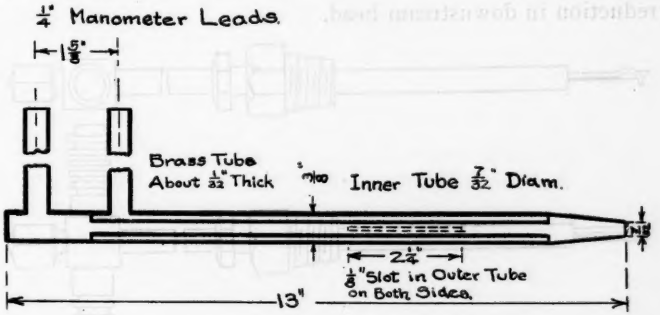



FIG. 9.—TAYLOR TUBE.

the impact and static tube in one case, the dynamic side communicating to the manometer through the inner tube and rear lead, and the static connection through long slots by way of the forward lead. These slots effectively eliminate aspiration troubles encountered in simple impact opening. Even better results are obtained by the use of four openings in the ⅜-inch tube, provided the width of slot is between .02-inch and .03-inch. The great length of this tube makes its use with high velocities attendant with some difficulties, but it gives excellent results with the velocities usually encountered.

Figure 10 (p. 1134) illustrates the tube designed by Prof. G. D. Gebhardt of the Armour Institute of Technology. The dynamic tube is a small tube bent on a small radius, and the static tube along side of it is a plain tube beveled to an angle 45 degrees. "In this device the dynamic and aspiration effects neutralize each other and only the true pressure is recorded. Further experiments are necessary to show whether any fixed angle is applicable to all velocities." *

* The Pitot Tube as a Steam Meter. Geo. F. Gebhardt, "Journal A. S. M. E.," Mid-November, 1909, page 1244.

This aspiration effect explains the unsatisfactory performance of the double tube with one opening upstream and the other downstream, an arrangement which should theoretically give a manometer reading equal to double the velocity head. Furthermore, the distortion of stream lines accounts for the reduction in downstream head.

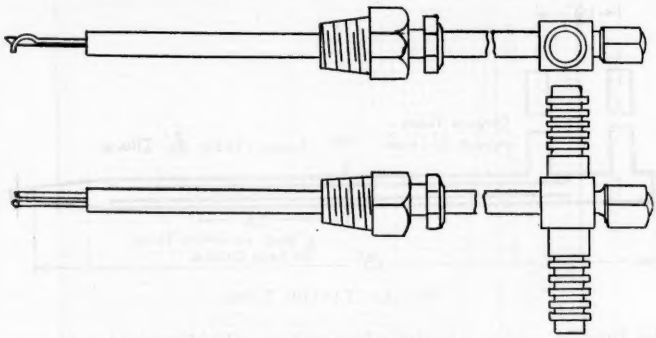


FIG. 10.—GEBHARDT TUBE.

The measurement of the velocity head of a current of air involves serious difficulties, since the velocity corresponding to an impact head of one inch of water is over 4,000 feet per minute. When the velocity exceeds that corresponding to one inch of water the measurements can be most simply made on an ordinary U-tube, with water or oil of known specific gravity as the measuring liquid. Readings can be secured with a probable precision of .01 inch, and the proportional error in the velocity computed therefrom will be but half the percentage error of the head reading. To eliminate double readings of level of liquid at bottom and top of the scale a special movable scale, graduated both ways from zero, may be applied. Then if the bore of the U-tube is uniform and the zero is adjusted to the no-pressure position of the water level, either the upper or lower level may be read and the true head recorded as double the observed head. If the scale be divided in the ratio of 2 : 1, then the head indicated above one side of the zero will be the true head.

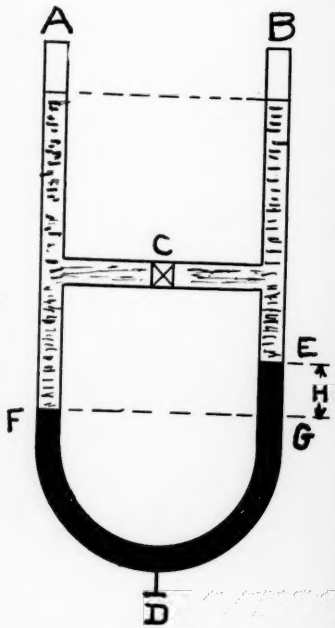



FIG. 11.

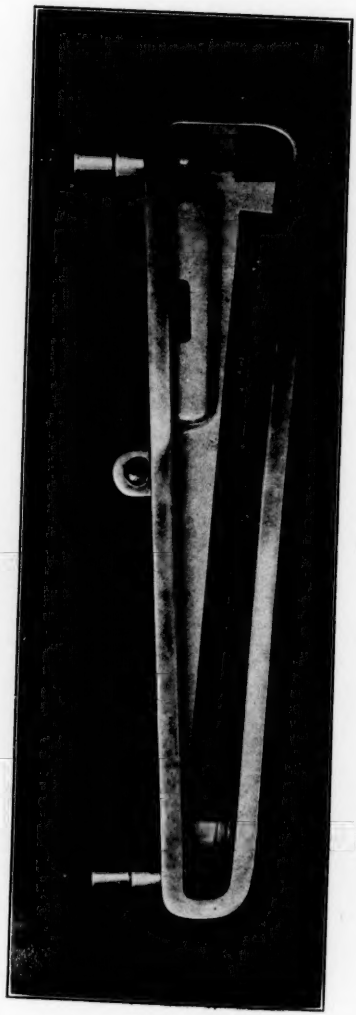


FIG. 12.

Magnification of the dynamic head is frequently accomplished by the use of liquids of specific gravity less than that of water. This method results in a very slight magnification and is scarcely practicable, as the lighter oils are usually quite volatile and must be replenished frequently.

The velocity corresponding to a head of one inch of water is approximately 4,030 feet per minute, and as the velocities encountered where measurements are desired are usually less than this amount, it is necessary to provide some means of accurately measuring a head when the simple U-tube with its scale precise to one or two hundredths of an inch is liable to introduce too great a percentage of error.

By the proper selection of liquids the differential gage may serve to give any desired magnification. The action of this gage is illustrated in Fig. 11. The static tube B and the dynamic tube A contain a small quantity of oil of specific gravity d, and below the oil in the U-tube from E to F is a quantity of water. Opening the valve at C by-passes the water leg when valve D is closed. The operation of this form of gage is as follows: Close valve C and open D, allowing the water column to be lifted. Then close D and open C to equalize the head of oil in both tubes. Repeat to insure perfect equalization of oil levels in A and B. When this has been accomplished, H, the head of water from E to G, is balanced by an equal head of oil in the leg A and the dynamic pressure, h. The dynamic pressure in inches of water will be

$$h = (1 - d)\, H.$$

If the specific gravity of the oil has been taken to be .9 the true head will thereby be multiplied by 10 in the gage.

The inclined U-tube, which in a commercial form is shown in Fig. 12, serves the purpose of magnification simply and satisfactorily. In the gage shown the slant is such that 0.1 inch on the scale is 0.01 inch of vertical difference, so that thousandths of an inch may be estimated with considerable accuracy. The fluid used is a light oil of density about the

same as water. The oil is sometimes colored red to give a plainer indication, and suffers practically no loss due to evaporation. A small spirit level is attached to the gage to insure the inclined tube having the slant intended.

## RELATION BETWEEN CENTER AND AVERAGE VELOCITY IN CIRCULAR DUCTS.

Considerable research has been made in the past to establish, by means of Pitot tube surveys, the relation between average and center velocity of water flowing in channels of various shapes, but no experiments to determine a like ratio in the case of air have come under the writer's notice. There are, however, available from several sources complete data of traverses in a few circular ducts which have been worked over to bring out this relation, in addition to the tests conducted at the Experiment Station.

In the absence of reliable data on the relation between center and average velocity earlier experimenters used several Pitot tubes distributed over the cross-section of the duct and connected each to a compartment of a multiple manometer which was assumed to indicate the average velocity head. This was the arrangement used by Naval Constructor Taylor in his "Experiments with Ventilating Fans and Pipes," reported at the thirteenth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 16 and 17, 1905. The object of these valuable experiments was to learn "more than was known about the capacities and efficiencies of fans such as are ordinarily used for ventilation purposes on U. S. Naval vessels and to obtain much needed information to enable systems of ventilation piping to be intelligently designed."

Figures 13, 14, 15 and 16 show a typical installation of Pitot tubes used by Mr. Taylor, consisting of nine or ten tubes distributed over the section of a 12-inch pipe in such a manner as to divide the area to be investigated into ten concentric rings of equal area and to place one tube at the mid-radius of each ring. Mr. Taylor recognized the fact that the

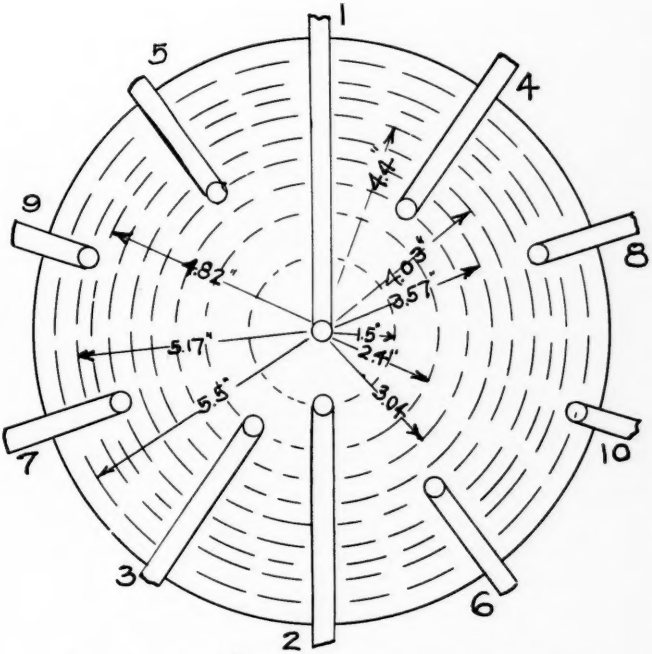



FIG. 13.—ARRANGEMENT OF 10 TUBES IN 12-INCH PIPE.

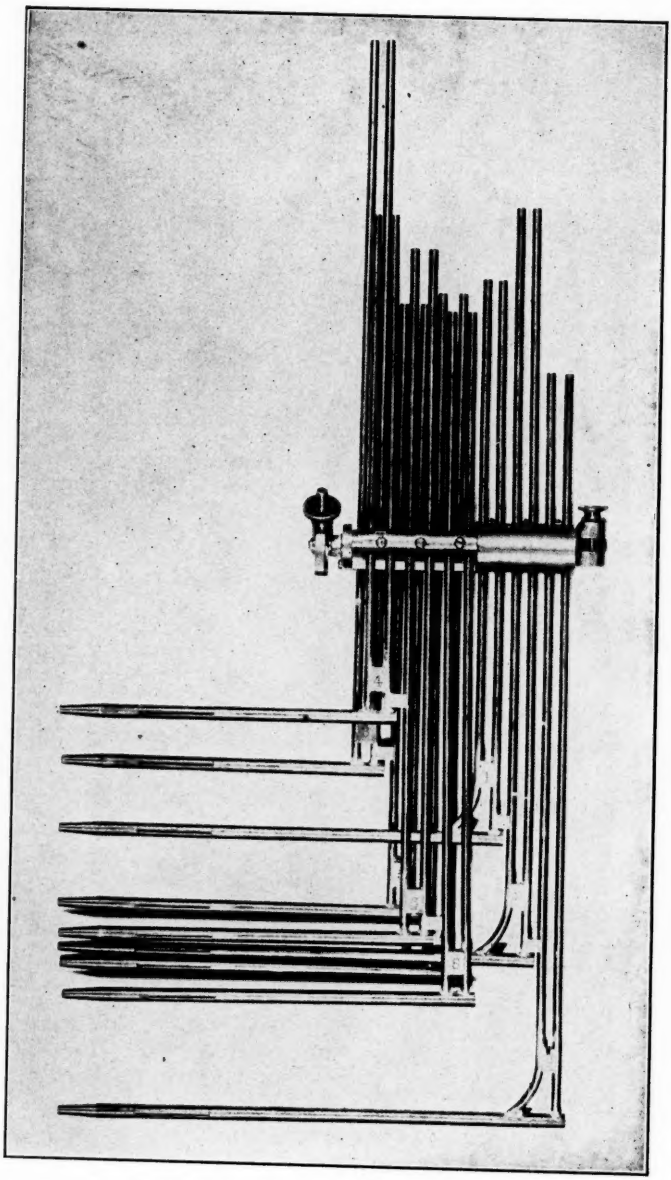


FIG. 14.—ADJUSTABLE PITOT TUBE SET.

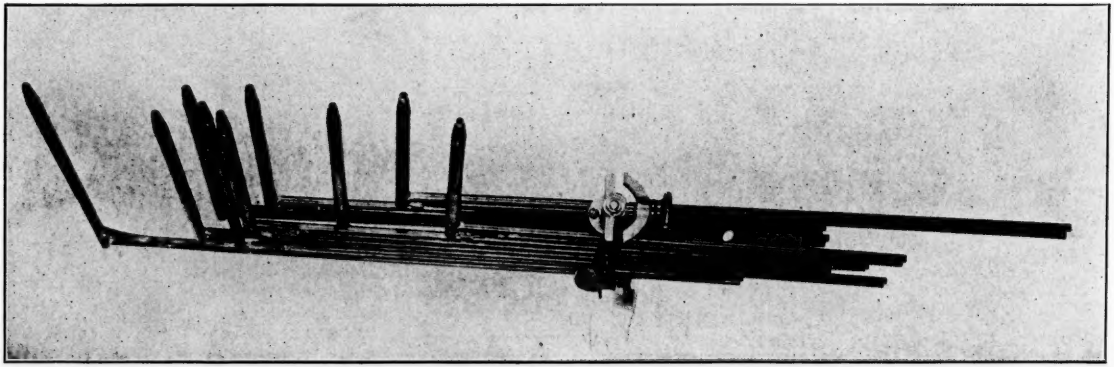


FIG. 15.—ADJUSTABLE PITOT TUBE SET.

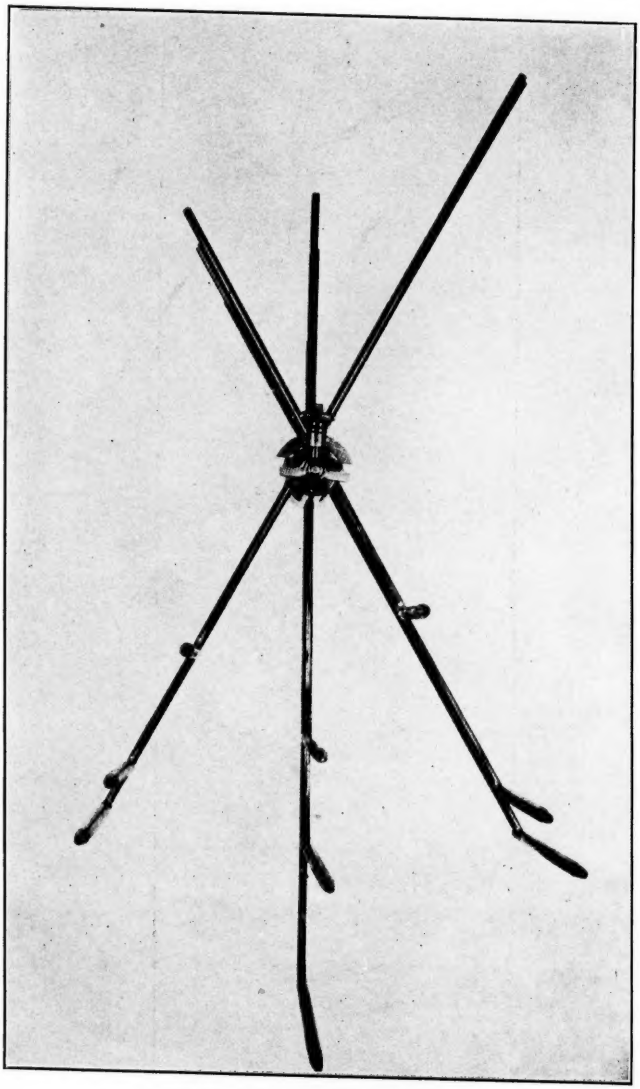


FIG. 16.—ADJUSTABLE PITOT TUBE SET.

velocity would not be uniform across the section, and hence determined to use not less than ten tubes communicating to a single manometer.

"This method is theoretically in error, since we assume the average velocity is that due to the average pressure. Suppose there are $n$ tubes used, that $V$ is the true average velocity, and that $X_1, X_2, X_3 \ldots . X_n$ are the differences between the true average velocity and the velocities of the $n$ points, some of the differences being positive and some negative, of course. Let $\triangle P$ denote the total excess pressure or head due to velocity, as measured by the manometer.

"Then

$$\triangle P = \text{Const.}\,[(V+X_1)^2 + (V+X_2)^2 + (V+X_3)^2 + \ldots . (V+X_n)^2]$$
$$= \text{Const.}\,[nV^2 + 2V(X_1 + X_2 + X_3 + \ldots . X_n) + X_1^2 + X_2^2 + X_3^2 + \ldots . X_n^2].$$

"Now, since $V$ is the average velocity,

$$X_1 + X_2 + X_3 + \ldots . X_n = 0,$$

so we have

$$\triangle P = \text{Const.}\,(nV^2 + X_1^2 + X_2^2 + X_3^2 + \ldots . X_n^2).$$

"If $\triangle P$ corresponded to average velocity only we should have $\triangle P = \text{Const.}\,(nV^2)$ only. Hence, when $X_1, X_2$, etc., have real values, *i. e.*, when there is turbulent motion in the pipe, the velocity shown by the instrument will be higher than the true average velocity. In practice, however, this error is not at all serious. For instance, if we make the rather extreme assumption that the actual velocity at half of the tubes is 10 per cent. above the average and at the other half 10 per cent. below the average, the error, by assuming the velocity to be that due to the average pressure, is but one-half of one per cent."

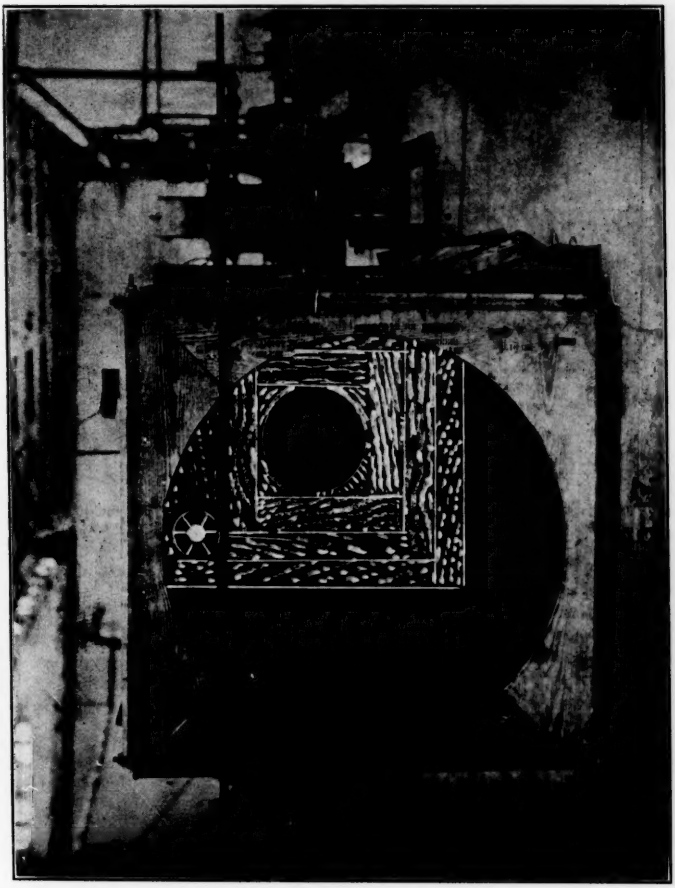
It will be shown from a number of tests that the variation across any section is considerably more than 10 per cent. from the mean, at moderate velocities, so that the single manometer indicates not the average velocity but, as was pointed

out in considering Prof. Rateau's work, the average energy of the stream.

In the experiments by Prof. Thomas, Pitot tube readings were taken on a vertical diameter at one-inch intervals from the center of a 16-inch pipe to five inches out, and at one-half inch intervals from this point to within one-half inch of the inner surface. The value of average square root of head was determined by giving a weight to each velocity (square root of velocity head) proportional to the area of the annular ring whose mean diameter was the diameter of the tube circle. These values are somewhat greater than those determined by the graphical method described in the "Journal of the Franklin Institute" for November, 1911. The results of the experiments on the traverse of the pipe are given in Table VI (p. 1139).

In testing the discharge from a turbine-driven fan, an airtight box 16 feet long and 5 feet square was constructed as shown in Fig. 17. Mid-length of the box a diaphragm was arranged to give various openings up to 21½ inches by two interchangeable plates, 12 inches and 21½ inches in diameter. The velocity head at the diaphragm was obtained by means of two Taylor Pitot tubes, mounted at right angles and movable to permit of horizontal and vertical traverses at the points shown in Fig. 18. The data and results are given in Table VII (page 1139). Particular attention is called to the fact that vertical and horizontal points equidistant from the center do not necessarily have the same velocity head, hence the necessity of both sets of observations. In one case the maximum head was not found at the center of the diaphragm.

The ratings by the writer were, as previously noted, preliminary to the test of an air heater. Two standard Taylor Pitot tubes were mounted in the 12½-inch circular duct, Fig. 18, in such a way as to enable one tube, $P_1$, to traverse the horizontal diameter and the other, $P_2$, to traverse the vertical diameter. These tubes had independent connections to the draft gage, D, which was so arranged with stop cocks as to permit of registering the velocity head from either tube, and



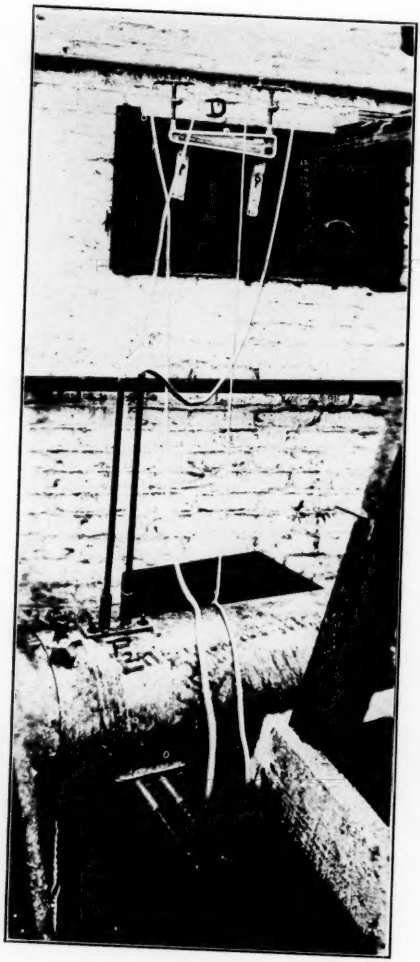



FIG. 18.

| Test No. | Barometer. Pounds per Sq. inch. | Relative Humidity. % | Temperature of Air - $t_a$. | Static Air Pressure Inches of water. | Absolute air pressure, pounds per square inch. | Density of dry air pounds per cubic foot. | Density of vapor, pounds per cubic foot. | Density of moist air, pounds per cubic foot. | TABLE VI Velocity head - Inches of water | | | | | | | | | | | Square root of head at Center. | Average Square Root of Head. | Ratio Average Velocity to Center Velocity. | Average Head in inches of Water. | Center Velocity - Feet per minute. | Pounds of Air Per Hour. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | Center of Pipe | Cm 2-inch diameter. | Cm 4-inch diameter. | Cm 6-inch diameter. | Cm 8-inch diameter. | Cm 10-inch diameter | Cm 11-inch diameter | Cm 12-inch diameter | Cm 13-inch diameter | Cm 14-inch diameter | Cm 15-inch diameter | | | | | | |
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 |
| 1 | 14.2315 | 63 | 82 | .430 | 14.2470 | .07327 | .00167 | .07432 | .150 | .150 | .148 | .143 | .138 | .136 | .133 | .131 | .128 | .117 | .100 | .3873 | .3573 | .922 | .1277 | 1560 | 8955 |
| 2 | 14.2315 | 63 | 83 | .655 | 14.2582 | .07314 | .00172 | .07422 | .230 | .230 | .228 | .222 | .216 | .210 | .205 | .200 | .190 | .175 | .145 | .4796 | .4408 | .919 | .1943 | 1932 | 11180 |
| 3 | 14.2315 | 63 | 84 | .930 | 14.2651 | .07300 | .00178 | .07412 | .336 | .336 | .330 | .320 | .310 | .300 | .294 | .285 | .277 | .255 | .208 | .5797 | .5294 | .913 | .2802 | 2331 | 13200 |
| 4 | 14.2315 | 63 | 85 | 1.085 | 14.2707 | .07287 | .00183 | .07402 | .385 | .385 | .383 | .370 | .362 | .352 | .345 | .335 | .322 | .295 | .240 | .6205 | .5696 | .918 | .3245 | 2501 | 14220 |
| 5 | 14.2315 | 63 | 85 | 1.200 | 14.2748 | .07287 | .00183 | .07402 | .460 | .455 | .448 | .435 | .425 | .410 | .405 | .390 | .375 | .350 | .290 | .6782 | .6184 | .912 | .3824 | 2729 | 15410 |
| 6 | 14.2315 | 63 | 87 | 1.520 | 14.2863 | .07261 | .00195 | .07384 | .552 | .548 | .538 | .520 | .516 | .500 | .488 | .475 | .450 | .420 | .355 | .7430 | .6802 | .915 | .4627 | 3000 | 16990 |
| 7 | 14.2266 | 63 | 895 | 1.765 | 14.2903 | .07220 | .00210 | .07352 | .640 | .632 | .622 | .605 | .592 | .572 | .562 | .538 | .515 | .475 | .400 | .8000 | .7273 | .909 | .5289 | 3234 | 18110 |
| 8 | 14.2266 | 63 | 91 | 2.070 | 14.3013 | .07207 | .00220 | .07345 | .710 | .705 | .685 | .670 | .650 | .627 | .610 | .595 | .575 | .525 | .455 | .8426 | .7654 | .908 | .5858 | 3414 | 19080 |
| 9 | 14.2266 | 63 | 90 | 1.150 | 14.2681 | .07221 | .00213 | .07355 | .385 | .385 | .378 | .365 | .357 | .348 | .340 | .332 | .320 | .292 | .248 | .6205 | .5689 | .917 | .3237 | 2512 | 14200 |
| 10 | 14.2266 | 63 | 85 | .460 | 14.2332 | .07287 | .00183 | .07402 | .125 | .125 | .123 | .120 | .117 | .114 | .112 | .109 | .102 | .089 | .080 | .3536 | .3256 | .921 | .1058 | 1428 | 8145 |

| No. | Barometer inches of Hg. | Relative Humidity %. | Temp. of Air | Static Air Pressure inches of Water. | TABLE VII. Velocity Head Inches of Water. 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | Average Square Root of Head. | Square Root of Head at Center. | Ratio Average Velocity to Center Velocity. | Average Head Inches of Water. | Center Velocity Feet per Minute. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| 1 | 30.10 | 61 | 74 | 4.53 | 3.62 | 4.58 | 4.62 | 4.60 | 4.18 | 4.20 | 4.46 | 4.62 | 4.64 | 4.52 | | | | | 2.076 | 2.149 | .966 | 4.31 | 6600 |
| 2 | 30.10 | 51 | 81 | 2.64 | 1.86 | 2.60 | 2.58 | 2.22 | 1.68 | 2.24 | 2.62 | 2.62 | 2.68 | 2.20 | | | | | 1.478 | 1.612 | .917 | 2.168 | 4735 |
| 3 | 30.10 | 50 | 845 | 1.27 | .68 | 1.14 | 1.14 | 1.14 | .74 | .86 | 1.14 | 1.14 | 1.16 | .80 | | | | | .946 | 1.068 | .889 | .695 | 2875 |
| 4 | 30.10 | 57 | 80 | 3.44 | 3.96 | 4.10 | 4.22 | 4.20 | 4.20 | 4.04 | 4.00 | 3.84 | 4.06 | 4.14 | 4.22 | 4.32 | 4.34 | 4.00 | 2.015 | 2.052 | .982 | 4.06 | 5810 |
| 5 | 30.11 | 48 | 83 | 2.06 | 2.32 | 2.40 | 2.44 | 2.44 | 2.40 | 2.40 | 2.34 | 2.26 | 2.32 | 2.42 | 2.44 | 2.44 | 2.40 | 2.36 | 1.535 | 1.562 | .983 | 2.36 | 4855 |
| 6 | 30.11 | 50 | 85 | .77 | .98 | 1.04 | 1.08 | 1.04 | 1.06 | 1.02 | 1.00 | .96 | 1.00 | 1.06 | 1.10 | 1.10 | 1.10 | 1.08 | 1.012 | 1.034 | .979 | 1.024 | 3245 |

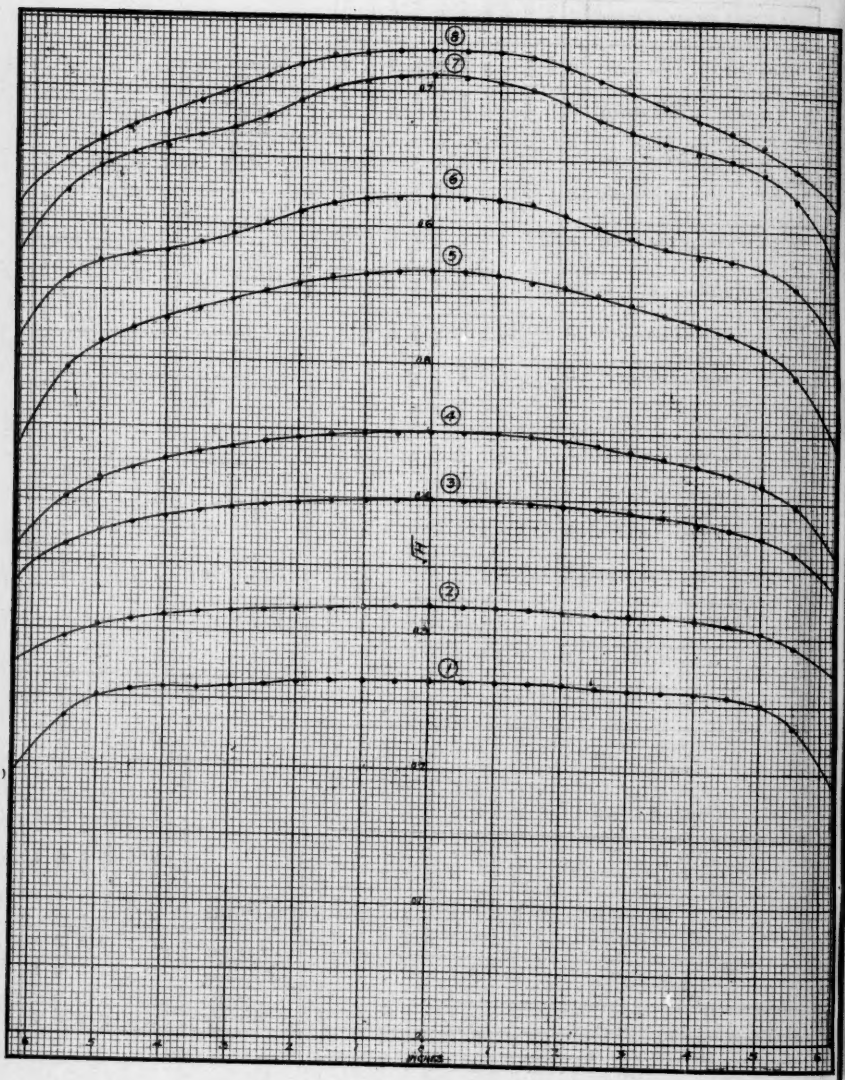



FIG. 19.

| Test Number. | Barometer, inches of Hg. | Relative Humidity, per cent. | Temperature of air, degrees F. | Static air pressure, inches of water. | Vapor pressure, inches of Hg. | Absolute air pressure, pounds per sq. inch. | Density of dry air, pounds per cubic foot. | Density of vapor, pounds per cubic foot. | Density of moist air, pounds per cubic foot. | Center of inlet pipe |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
| 1 | 30.13 | 56.27 | 46.5 | 0.66 | 0.18 | 14.67 | 0.07824 | 0.00036 | 0.07860 | 0.0703 |
| 2 | 29.61 | 54.47 | 43.5 | 0.74 | 0.16 | 14.48 | 0.07769 | 0.00027 | 0.07796 | 0.1030 |
| 3 | 29.98 | 65.48 | 44.0 | 1.06 | 0.19 | 14.67 | 0.07863 | 0.00028 | 0.07891 | 0.1602 |
| 4 | 29.96 | 47.10 | 46.0 | 1.50 | 0.15 | 14.64 | 0.07816 | 0.00024 | 0.07840 | 0.2018 |
| 5 | 29.92 | 42.51 | 50.0 | 2.32 | 0.15 | 14.70 | 0.07786 | 0.00023 | 0.07809 | 0.3221 |
| 6 | 29.88 | 38.34 | 51.0 | 2.84 | 0.14 | 14.71 | 0.07776 | 0.00021 | 0.07797 | 0.3881 |
| 7 | 29.88 | 39.03 | 56.0 | 3.70 | 0.18 | 14.72 | 0.07706 | 0.00021 | 0.07727 | 0.5030 |
| 7-a | 30.20 | 55.35 | 45.0 | 3.93 | 0.17 | 14.89 | 0.07965 | 0.00022 | 0.07987 | 0.5101 |
| 8 | 30.20 | 54.93 | 45.0 | 4.05 | 0.16 | 14.90 | 0.07970 | 0.00023 | 0.07993 | 0.5400 |
| 8-a | 30.01 | 79.00 | 47.8 | 3.77 | 0.25 | 14.75 | 0.07846 | 0.00040 | 0.07886 | 0.5284 |

TABLE VIII.—PITOT TUBE RATING.

| | Velocity Head --- Inches of Water. | | | | | | |
|---|---|---|---|---|---|---|---|
| Center of inlet pipe | On 1-inch diameter. | On 2-inch diameter. | On 3-inch diameter. | On 4-inch diameter. | On 5-inch diameter. | On 6-inch diameter. | On 7-inch diameter. |
| 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
| 0.07032 | 0.07096 | 0.07037 | 0.07008 | 0.06929 | 0.06833 | 0.06763 | 0.06741 |
| 0.1030 | 0.1023 | 0.1028 | 0.1021 | 0.1010 | 0.1003 | 0.0998 | 0.0989 |
| 0.1602 | 0.1593 | 0.1592 | 0.1581 | 0.1566 | 0.1549 | 0.1532 | 0.1508 |
| 0.2018 | 0.2010 | 0.2008 | 0.1998 | 0.1969 | 0.1948 | 0.1915 | 0.1870 |
| 0.3221 | 0.3215 | 0.3189 | 0.3153 | 0.3101 | 0.3045 | 0.2958 | 0.2900 |
| 0.3881 | 0.3867 | 0.3853 | 0.3808 | 0.3733 | 0.3625 | 0.3529 | 0.3436 |
| 0.5030 | 0.5033 | 0.4988 | 0.4885 | 0.4753 | 0.4649 | 0.4464 | 0.4303 |
| 0.5101 | 0.5068 | 0.5023 | 0.4979 | 0.4856 | 0.4734 | 0.4577 | 0.4456 |
| 0.5400 | 0.5411 | 0.5364 | 0.5300 | 0.5205 | 0.5070 | 0.4870 | 0.4771 |
| 0.5284 | 0.5251 | 0.5251 | 0.5210 | 0.5116 | 0.4977 | 0.4842 | 0.4716 |

TABLE VIII.—PITOT TUBE RATING.

| | Velocity Head --- Inches of Water. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Center of inlet pipe | On 1-inch diameter. | On 2-inch diameter. | On 3-inch diameter. | On 4-inch diameter. | On 5-inch diameter. | On 6-inch diameter. | On 7-inch diameter. | On 8-inch diameter. | On 9-inch diameter. |
| 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 |
| .07032 | 0.07096 | 0.07037 | 0.07008 | 0.06929 | 0.06833 | 0.06763 | 0.06741 | 0.06688 | 0.06579 |
| .1030 | 0.1023 | 0.1028 | 0.1021 | 0.1010 | 0.1003 | 0.0998 | 0.0989 | 0.0975 | 0.0954 |
| .1602 | 0.1593 | 0.1592 | 0.1581 | 0.1566 | 0.1549 | 0.1532 | 0.1508 | 0.1477 | 0.1443 |
| .2018 | 0.2010 | 0.2008 | 0.1998 | 0.1969 | 0.1948 | 0.1915 | 0.1870 | 0.1825 | 0.1773 |
| .3221 | 0.3215 | 0.3189 | 0.3153 | 0.3101 | 0.3045 | 0.2958 | 0.2900 | 0.2816 | 0.2740 |
| .3881 | 0.3867 | 0.3853 | 0.3808 | 0.3733 | 0.3625 | 0.3529 | 0.3436 | 0.3369 | 0.3343 |
| .5030 | 0.5033 | 0.4988 | 0.4885 | 0.4753 | 0.4649 | 0.4464 | 0.4303 | 0.4214 | 0.4191 |
| .5101 | 0.5068 | 0.5023 | 0.4979 | 0.4856 | 0.4734 | 0.4577 | 0.4456 | 0.4375 | 0.4275 |
| .5400 | 0.5411 | 0.5364 | 0.5300 | 0.5205 | 0.5070 | 0.4870 | 0.4771 | 0.4658 | 0.4589 |
| .5284 | 0.5251 | 0.5251 | 0.5210 | 0.5116 | 0.4977 | 0.4842 | 0.4716 | 0.4612 | 0.4487 |

| On 9-inch diameter. | On 10-inch diameter. | On 11-inch diameter. | On 12-inch diameter. | Square root of head at center. | Average square root of head. | Ratio average velocity to center velocity. | Center velocity, feet per minute. |
|---|---|---|---|---|---|---|---|
| 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 |
| 0.06579 | 0.06313 | 0.05509 | 0.04264 | 0.2663 | 0.2472 | 0.9282 | 1042.0 |
| 0.0954 | 0.0922 | 0.0855 | 0.0748 | 0.3209 | 0.3037 | 0.9464 | 1260.8 |
| 0.1443 | 0.1397 | 0.1317 | 0.1190 | 0.4003 | 0.3766 | 0.9407 | 1563.2 |
| 0.1773 | 0.1702 | 0.1584 | 0.1446 | 0.4492 | 0.4179 | 0.9303 | 1759.9 |
| 0.2740 | 0.2636 | 0.2427 | 0.2111 | 0.5675 | 0.5183 | 0.9133 | 2227.8 |
| 0.3343 | 0.3278 | 0.3117 | 0.2846 | 0.6230 | 0.5764 | 0.9252 | 2447.6 |
| 0.4191 | 0.4069 | 0.3814 | 0.3452 | 0.7092 | 0.6440 | ----- | ------ |
| 0.4275 | 0.4215 | 0.4033 | 0.3552 | 0.7142 | 0.6541 | 0.9132 | 2789.9 |
| 0.4589 | 0.4496 | 0.4264 | 0.3931 | 0.7349 | 0.6773 | ----- | ------ |
| 0.4487 | 0.4295 | 0.3866 | 0.3648 | 0.7269 | 0.6644 | 0.9165 | 2843.6 |

the gage was fitted with an additional pair of equalizing cocks which could be opened to the atmosphere between tests in order to check the zero reading of the draft gage. The single divisions on the scale of the draft gage were 0.01 inch, and 0.001 inch could be estimated with a probable precision of ± 0.001 inch. Half-inch intervals were etched on one of the ⅛-inch pipes connecting to the Pitot tube.

The procedure was as follows: With the fan speed constant a traverse was made with the horizontal tube, a reading taken at each of twenty-three points across the pipe, and at the same points on the return. The same was followed by a traverse with the vertical tube and each set repeated twice, making a total of 276 observations in each set. The fan speed was then increased and the procedure repeated for the additional points covering the range of fan capacity. The data from the tests are given in Table VIII and the variations of velocity across the cross-section are shown in Fig. 19, in which the ordinates are $\sqrt{H}$ and the abscissae are the corresponding positions in the duct. From an extrapolation of these curves to a diameter of 12 inches the values in column 23 of Table VIII are obtained. In arriving at the average square root of the head as tabulated in column 25, the square root of the head at the extremity of each diameter was given a weight in proportion to the area of the annular ring between the circumference of circles drawn halfway between the given diameter and the two adjacent diameters. These relative weights were as follows:

| | |
|---|---|
| Center, | 0.16 |
| 1-inch diameter, | 1.28 |
| 2-inch diameter, | 2.56 |
| 3-inch diameter, | 3.84 |
| 4-inch diameter, | 5.12 |
| 5-inch diameter, | 6.40 |
| 6-inch diameter, | 7.68 |
| 7-inch diameter, | 8.96 |
| 8-inch diameter, | 10.24 |

| | |
|---|---|
| 9-inch diameter, . . . . | 11.52 |
| 10-inch diameter, . . . . | 12.80 |
| 11-inch diameter, . . . . | 14.08 |
| 12-inch diameter, . . . . | 15.36 |
| Total . . . . . | 100.00 |

The irregularity in the shape of the curves in Fig. 19 at high velocities is to be charged to the disturbing effect of an elbow in the line. This elbow was eight diameters away from the point at which the Pitot tube readings were taken, but its effect was noticeable at velocities over 2,000 feet per minute.

It has been suggested that the mean head could be determined by a single tube placed at such a point as to intersect the curves at the mean height. This particular point would be at a distance of $5\frac{1}{4}$ inches from the center of the $12\frac{1}{2}$-inch duct. There are two serious objections to this method: first, readings on a horizontal and on a vertical traverse are never the same except at the points nearest the center and it would be impossible to locate the mean point; and second, the velocity is changing rapidly in this region and the greatest care is necessary in adjusting the tube to this point, even if the first difficulty could be overcome. The great advantage of the single central tube with its proper correction factors is brought out by the inspection of the central region of the curve, Fig. 19, which shows clearly that for a half-inch to either side of the center the velocity head remains practically constant, thereby giving a region in which settings of the central tube may readily be made.

The variation of the ratios $\frac{\text{Average Velocity}}{\text{Center Velocity}}$ from these tests is plotted in Fig. 20, and with it the corresponding curve from Prof. Thomas' tests. The agreement between them is very satisfactory.

An analysis of the three sets of data show that there may easily be cases where half the points in a cross-section have velocities 10 per cent. above the average, and hence a

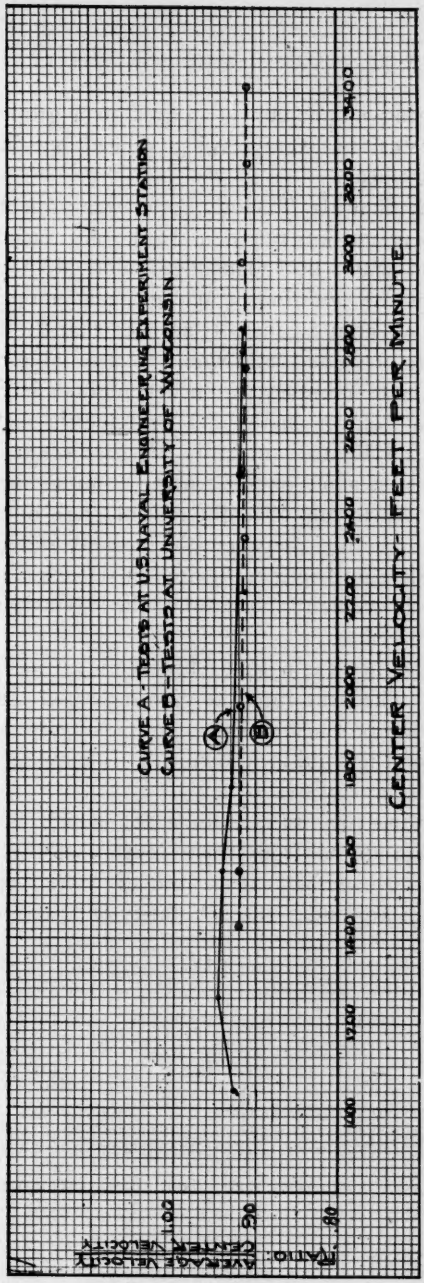



FIG. 20.—PITOT TUBE RATING.

single averaging manometer connected to a group of tubes will not register the average velocity head, but instead, the average energy head.

At the time that Mr. Taylor's paper was presented a considerable discussion was raised as to the accuracy of the instrument, and while no independent method of measuring the air except the notoriously inaccurate anemometer could be devised, it was quite generally acceded that the Pitot tube when properly made and used would give reliable results.

Since that time, however, two other methods have become available, the Venturi meter and the electric meter. The principle of the Venturi meter is the gradual reduction of the flow section to about one-fourth the nominal size, and the measurement by means of a differential gage of the difference of static pressures between the normal and contracted section. Neglecting friction, the drop in static head measures the increase in velocity head, and the discharge may be directly calculated.

The electric meter is a small heater unit installed in the line of flow. The heat input to give a desired temperature rise is a measure of the weight of gas passing, provided the specific heat of the fluid is known. This may be taken for air at normal temperatures as .2375.

The diagram, Fig. 21, the data for which is given in Table IX, reproduced from the "Journal of the Franklin Institute" for November, 1911, proves more conclusively than any other argument the reliability of any of the three

TABLE IX.

| Test No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|
| Pounds of air per hour : | | | | | | | | | | |
| Pitot tube..... | 8,960 | 10,520 | 13,190 | 14,040 | 15,440 | 17,000 | 18,120 | 19,160 | 13,970 | 8,150 |
| Venturi meter | 8,250 | 10,400 | 12,320 | 13,600 | 14,780 | 16,870 | 17,560 | 19,000 | 14,110 | 7,130 |
| Electric meter | 8,550 | 10,790 | 12,670 | 14,220 | 15,230 | 17,110 | 17,950 | 19,480 | 14,100 | 7,580 |
| Speed of fan, r.p.m ..... | 450 | 565 | 680 | 740 | 800 | 897 | 956 | 1,025 | 740 | 408 |

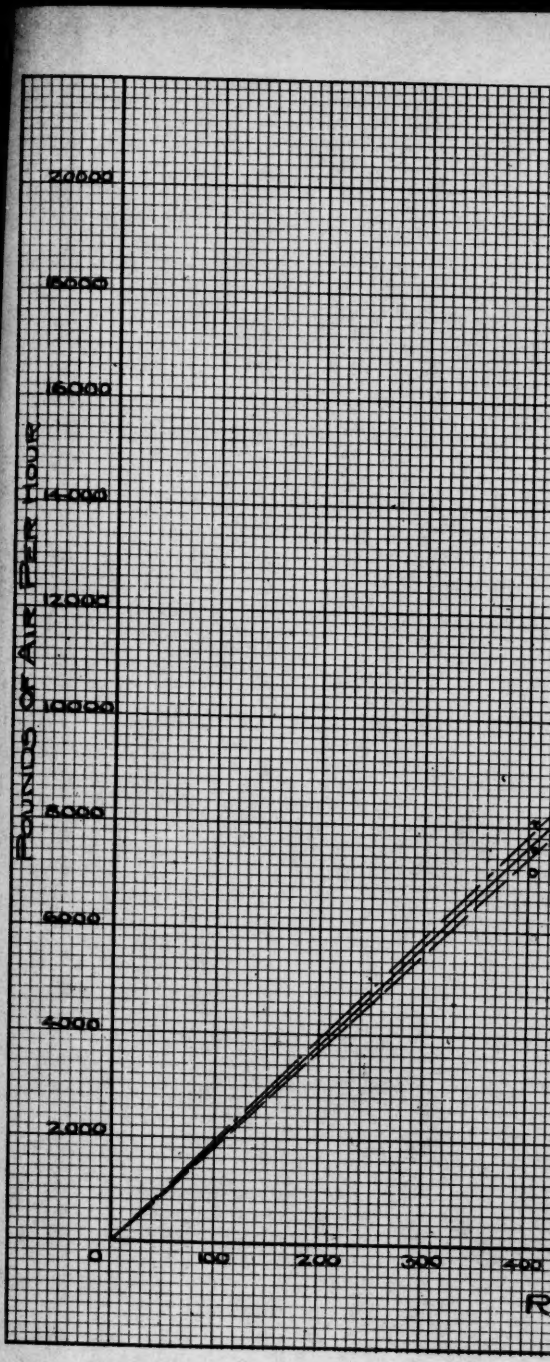
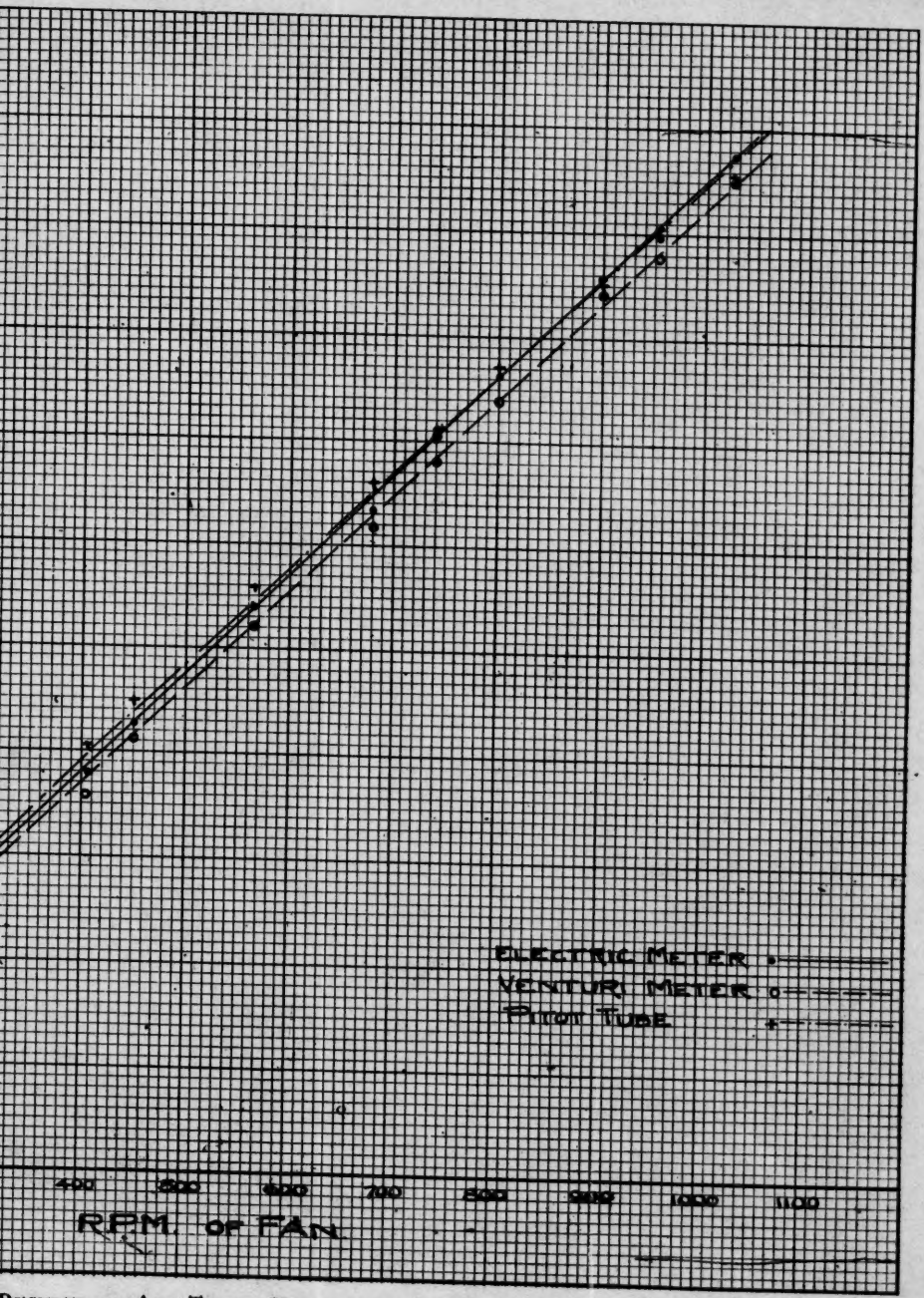



FIG. 21.—RESULTS




RESULTS OF AIR TESTS, UNIVERSITY OF WISCONSIN.

methods. This diagram represents Prof. Thomas' tests on the three forms of test apparatus installed in the same duct in the following order: fan, electric meter, Pitot tube, Venturi meter. The results from the electric meter and Pitot tube are generally higher than those from the Venturi meter, but the first two show excellent agreement between themselves.

## SUMMARY.

To summarize: The Pitot tube is a simple and reliable instrument for measuring the flow of air under pressure or into the atmosphere;

A single tube at the center of a circular duct will have a coefficient of mean velocity of from .91 to .94, depending on the size of duct and velocity of the air;

In a variable current an averaging manometer shows the energy head and not the velocity head;

Where the air carries a large percentage of moisture the usual caculations for dry air are apt to be considerably in error;

For velocities below 10,000 feet per minute the approximate formula for velocity is within the limits of accuracy of the data;

For velocities below 4,000 feet per minute some form of differential or multiplying gage is required to obtain the proper precision in velocity head.

# U. S. S. *ARKANSAS.*

## CONTRACT TRIALS PERFORMANCE.

By Henderson B. Gregory, Associate.

The U. S. S. *Arkansas* (Battleship No. 33) was built under contract by the New York Shipbuilding Company, of Camden, N. J., under an Act of Congress approved March 3, 1909, which authorized the construction of two battleships, the *Wyoming* being the sister ship. The contract price was $4,675,000.00, and time of completion thirty-two months from date of contract, September 25, 1909. The designed speed was 20.5 knots, at 26,000 tons displacement, and with the main engines (Parsons turbines, driving four screws) developing 28,000 S.H.P.

The hull and machinery are in general similar to the *Wyoming*, built by William Cramp & Sons, of Philadelphia, Pa., a description of which appeared in the August, 1912, number of the Journal of the American Society of Naval Engineers, and which applies to the *Arkansas* in all principal particulars, the condensing apparatus excepted. The latter is of the usual design fitted with Parsons vacuum augmenter, instead of the Weir uniflux condensers and dual air pumps as described for the *Wyoming*. All other differences are minor in character and need no reference here.

### TRIALS.

The contract required the following trials, five in all:

(*a*) A progressive trial over a measured-mile course for standardizing the screws.

(*b*) A full-speed trial of four hours duration in the open sea at the highest speed obtainable, with an average air pressure in the ash pits not exceeding 2 inches of water, and a steam pressure of not over 175 pounds per square inch above the atmosphere at the M.H.P. turbine.

(*c*) An endurance and coal-and-water-consumption trial of

twenty-four hours duration in the open sea, at as nearly as possible a uniform speed of 19 knots, the average not to fall below that figure. The trial to be conducted as nearly as possible under cruising conditions.

(*d*) An endurance and coal-and-water-consumption trial of twenty-four hours duration, at as nearly as possible an average speed of 12 knots.

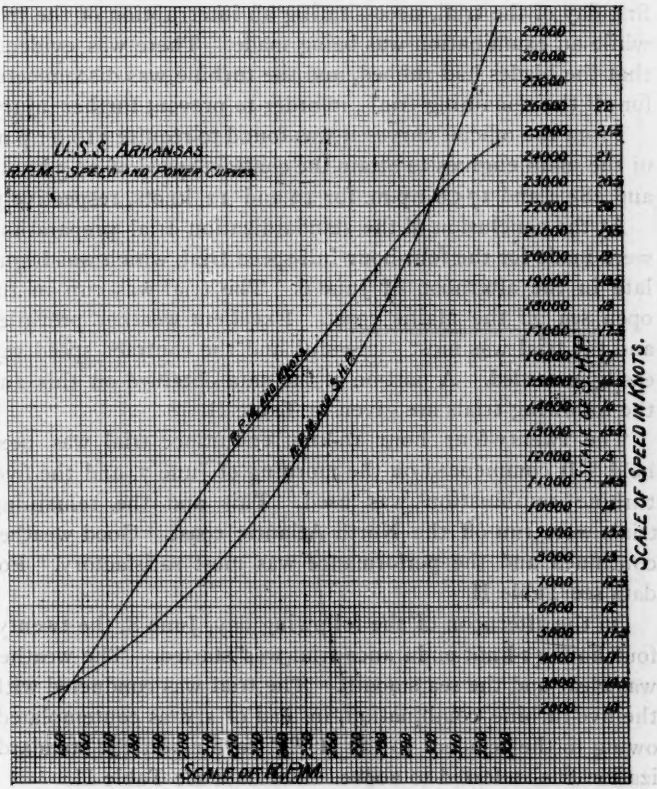



FIG. 1.

(*e*) A trial of two hours duration at the highest speed obtainable, burning coal and fuel oil in combination.

The standardization trial was run on the measured-mile course, at Rockland, Maine, on June 4 and 5, 1912. The

weather was favorable throughout the trial. In all, twenty-four runs were made over the course at various speeds, but the data from twenty-three of the runs only were used in compiling results and plotting the curves shown in Figure 1, run No. 7 being thrown out. Table I contains a portion of the data obtained.

Trouble was detected in the H.P.C. turbine early on the first day of the trial, necessitating an interruption of the trial while an examination was being made. There was evidence that the blades had rubbed, and the turbine was disconnected for all the remaining trials, in order to prevent further injury.

From the official curves it was found to require 310.9 r.p.m. of the main engines to attain the contract speed of 20.5 knots, and 284.4 and 175.8 r.p.m. for 19 and 12 knots, respectively.

At the conclusion of the standardization trial preparations were made for the four-hour full-speed trial, which was begun late in the afternoon of June 6. The trial was run in the open sea off the Maine coast. Excellent weather prevailed and the trial was most successful and the contract speed was easily exceeded. A portion of the data obtained on this and the following trials are given in Table II.

The twenty-four hour 19-knot endurance trial was next held. It commenced on the morning of June 7, and the five-turbine combination was used. This and the remaining trials were run off the North Atlantic coast. Good weather continued and the performance was most satisfactory. For data see Table II.

On the afternoon of the following day, June 8, the twenty-four hour 12-knot endurance trial was started. The weather was clear and the sea smooth. The trial was conducted with the five-turbine combination, instead of six as contemplated, owing to the accident to the H.P.C. turbine on the standardization trial referred to above. For data see Table II.

The two-hour coal-and-fuel-oil-burning trial followed on the afternoon of June 9, under favorable weather conditions. The performance of the oil-burning installation and the general results of the trial were entirely satisfactory. For data see Table II.

TABLE I.—STANDARDIZATION TRIAL DATA U. S. S. "ARKANSAS."

TABLE I.—STANDARDIZATION TRIAL DATA U. S. S. "ARKANSAS."

Rockland, Maine. June 4 and 5, 1912.

| Number of run..... | 1 | 2 | 3 | 4 | 5 | 6 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Time on course, min. sec.. | 6 8.4 | 5 41.35 | 5 50 | 4 42.73 | 4 38.9 | 4 45.93 | 3 44.83 | 3 41.63 | 3 43.5 | 3 26.5 | 3 30.5 | 3 27.17 | 3 14.5 | 3 13.23 | 3 15.1 | 2 50 | 2 54 | 2 49.87 | 2 52.53 | 2 47.77 | 2 52.63 | 2 47.5 | 2 50.67 |
| Revs. to go 1 knot: | | | | | | | | | | | | | | | | | | | | | | | |
| Shaft No. 1..... | 809.4 | 735.9 | 785.8 | 753.5 | 776.3 | 770.1 | 831.6 | 808.3 | 839.3 | 818.2 | 849.8 | 819.2 | 859.4 | 835.7 | 858.4 | 901.1 | 923.1 | 900.7 | 940.3 | 910.1 | 937.9 | 913.8 | 936.8 |
| No. 2..... | 1,070.8 | 000.3 | 1,037.6 | 994.5 | 998.8 | 1,008.3 | 1,051 | 1,020.5 | 1,040.6 | 1,017.6 | 1,044.2 | 1,017.9 | 1,024.6 | 1,005.1 | 1,027.9 | 924.1 | 942.9 | 920.8 | 954.6 | 927.8 | 952.2 | 928.9 | 948.7 |
| No. 3..... | 1,034.3 | 950.3 | 994.1 | 959 | 961.7 | 976.5 | 831 | 848.7 | 865 | 848.7 | 870.2 | 850.2 | 876.7 | 860.8 | 878.4 | 910.4 | 931 | 908.6 | 940.7 | 910.5 | 938.8 | 914.1 | 933.1 |
| No. 4..... | 820.2 | 747.8 | 791.6 | 765 | 779.1 | 782.5 | 833.9 | 815 | 838.7 | 824.5 | 849.2 | 827.2 | 859.4 | 842.5 | 857.9 | 901.3 | 924.7 | 902.1 | 938.2 | 908 | 937.8 | 914.2 | 937 |
| Average..... | 933.7 | 856.1 | 902.3 | 868 | 879 | 884.5 | 886.9 | 873.2 | 896 | 877.3 | 903.4 | 878.7 | 905.1 | 886.1 | 905.7 | 909.3 | 930.5 | 908.1 | 943.5 | 914.2 | 941.7 | 917.8 | 939 |
| Revs. per minute: | | | | | | | | | | | | | | | | | | | | | | | |
| Shaft No. 1..... | 131.95 | 129.56 | 134.7 | 159.98 | 167.04 | 161.59 | 221.99 | 218.89 | 225.46 | 237.68 | 242.33 | 237.34 | 265.32 | 259.61 | 264.06 | 318.04 | 318.4 | 318.16 | 327.07 | 325.42 | 325.93 | 327.43 | 329.18 |
| No. 2..... | 174.67 | 174.34 | 177.88 | 211.15 | 214.91 | 211.56 | 280.58 | 276.37 | 279.53 | 295.6 | 297.78 | 294.89 | 316.3 | 312.21 | 316.2 | 326.15 | 325.23 | 325.26 | 332.04 | 331.74 | 330.9 | 332.83 | 333.35 |
| No. 3..... | 168.66 | 167.25 | 170.42 | 203.61 | 206.92 | 204.9 | 221.84 | 229.83 | 232.38 | 246.53 | 248.15 | 246.32 | 270.66 | 267.39 | 270.21 | 321.32 | 321.13 | 320.97 | 327.2 | 325.57 | 326.24 | 327.52 | 327.87 |
| No. 4..... | 133.68 | 131.59 | 135.7 | 162.4 | 167.64 | 164.27 | 222.61 | 220.71 | 225.31 | 239.5 | 242.16 | 239.65 | 265.3 | 261.71 | 263.89 | 318.09 | 318.95 | 318.67 | 326.31 | 324.68 | 325.89 | 327.57 | 329.26 |
| Average..... | 152.24 | 150.69 | 154.68 | 184.29 | 189.13 | 185.59 | 236.76 | 236 45 | 240.68 | 254.83 | 257.61 | 254.55 | 279.4 | 275.23 | 278.59 | 320.91 | 320.68 | 320.77 | 328.16 | 326.68 | 327.25 | 328.84 | 329.92 |
| Speed in knots ..... | 9.772 | 10.546 | 10.286 | 12.734 | 12.908 | 12.592 | 16.014 | 16.245 | 16.107 | 17.433 | 17.102 | 17.375 | 18.509 | 18.634 | 18.452 | 21.176 | 20.69 | 21.189 | 20.87 | 21.454 | 20.857 | 21.493 | 21.09 |
| S.H.P., Shaft No. 1.. | 237 | 220 | 249 | 528 | 576 | 566 | 1,798 | 1,620 | 1,646 | 2,139 | 2,193 | 2,159 | 2,971 | 2,920 | 2,957 | 6,219 | 6,304 | 6,300 | 6,705 | 6,736 | 6,779 | 6,909 | 6,978 |
| No. 2.. | 1,275 | 1,255 | 1,245 | 2,133 | 2,214 | 2,158 | 5,078 | 4,905 | 4,892 | 6,001 | 6,105 | 6,105 | 7,148 | 7,119 | 7,194 | 7,306 | 7,383 | 7,351 | 7,903 | 7,779 | 7,776 | 7,905 | 8,034 |
| No. 3.. | 983 | 1,045 | 1,065 | 1,883 | 1,852 | 1,895 | 1,797 | 2,241 | 2,196 | 2,503 | 2,692 | 2,795 | 3,613 | 3,663 | 3,621 | 6,940 | 6,840 | 6,853 | 7,296 | 7,276 | 7,178 | 7,320 | 7,279 |
| No. 4.. | 241 | 224 | 251 | 536 | 578 | 575 | 1,659 | 1,678 | 1,712 | 2,514 | 2,288 | 2,253 | 3,078 | 3,036 | 3,061 | 6,219 | 6,315 | 6,309 | 6,689 | 6,721 | 6,778 | 6,912 | 6,980 |
| Total, all shafts..... | 2,736 | 2,774 | 2,810 | 5,080 | 5,220 | 5,194 | 10,332 | 10,443 | 10,446 | 13,157 | 13,278 | 13,312 | 16,810 | 16,738 | 16,833 | 26,683 | 26,842 | 26,813 | 28,594 | 28,513 | 28,511 | 29,045 | 29,271 |
| Turbines used..... | H.P.C., I.P.C., M.H.P. & L. P. ahead (6). | | | | | | I.P.C., M.H.P. and L.P. ahead (5). | | | | | | | | | M.H.P. and L.P. ahead (4). | | | | | | | |

TABLE II.—U. S. S. "ARKANSAS."—TRIAL DATA.

| ..... | 4-hour full-power trial. | 24-hour 19-knot endurance trial. | 24-hour 12-knot endurance trial. | 2-hour coal-and-oil-burning trial. |
|---|---|---|---|---|
| Date of trial | June 6, 1912. | June 7–8, 1912. | June 8–9, 1912. | June 9, 1912. |
| Speed, in knots | 21.05 | 19.41 | 11.993 | 20.989 |
| Slip of propellers, per cent. of own speed: | | | | |
| Propeller No. 1 | 18.53 | 13.80 | 9.34 | 17.91 |
| No. 2 | 20.75 | 25.09 | 23.29 | 21.01 |
| No. 3 | 20.43 | 17.19 | 20.97 | 20.23 |
| No. 4 | 18.63 | 13.16 | 6.18 | 18.42 |
| Mean, all propellers | 19.59 | 17.31 | 14.94 | 19.39 |
| Draught, mean on trial, feet and inches | 27–11½ | 27–11¾ | 28–02 1/16 | 28–00 11/16 |
| Displacement corresponding, tons | 25,546 | 25,538 | 25,768 | 25,657 |
| Number of turbines used | 4 | 5 | 5 | 4 |
| boilers used | 12 | 12 | 8 | 12 |
| Grate surface used, square feet | 1,428 | 1,428 | 952 | 1,428 |
| Heating surface used, square feet | 64,234 | 64,234 | 42,822 | 64,234 |
| Number of oil burners in operation | None. | None. | None. | All. |
| Pressures (average): | | | | |
| Main steam at boilers, pounds gage | 215 | 215 | 206 | 215 |
| engine rooms, pounds gage | 199.5 | 208.2 | 204.3 | 195.5 |
| In steam belt, H.P.C. turbine, pounds absolute | ..... | H.P.C. turbine | disconnected. | ..... |
| I.P.C. turbine, pounds absolute | 13 | 190.9 | 62.9 | 15 |
| M.H.P. turbine, pounds absolute | 129.8 | 94.6 | 21.05 | 126.5 |
| L.P. turbine, pounds absolute | 15 | 11.2 | 3.55 | 13.75 |
| Vacuum in main condensers, inches of mercury | 28.55 | 28.87 | 29.18 | 28.1 |
| Auxiliary exhaust in feed heater, pounds gage | 1.76 | 2.53 | —6.76 | ..... |
| Forced-lubrication system, pounds gage | 13.7 | 16.3 | 15.2 | ..... |
| Air pressure in firerooms, inches of water | 1.08 | 0.597 | ..... | ..... |
| Temperatures, degrees F. (average): | | | | |
| Main injection | 49.5 | 50.1 | 57.8 | ..... |
| outboard delivery | 80.5 | 74.1 | 69.9 | ..... |
| Feed water | 155.5 | 166.1 | 183.4 | ..... |
| Engine rooms, working platform | 92.5 | 93.2 | 92.6 | ..... |
| Firerooms, working level | 84.6 | 91.2 | 93.3 | ..... |
| Outside air | ..... | 52 | ..... | ..... |
| Smokepipes | ..... | 486.8 | 347.5 | ..... |
| Revolutions or double strokes per minute (average): | | | | |
| Shaft No. 1 | 319.6 | 278.5 | 163.6 | 316.2 |
| No. 2 | 328.5 | 320.5 | 193.4 | 328.6 |
| No. 3 | 327.2 | 289.9 | 187.7 | 325.4 |
| No. 4 | 320 | 276.4 | 158.1 | 318.2 |
| Mean, all shafts | 323.8 | 291.3 | 175.7 | 322.1 |
| Main air pumps | 22 | 20 | 16.7 | ..... |
| circulating pumps | 600.5 | 568.6 | 512 | ..... |
| feed pumps | 25.1 | 19.7 | 15.3 | ..... |
| Forced-draft blowers | 667.3 | 654.6 | ..... | ..... |
| Horsepower, S.H.P. (average): | | | | |
| Shaft No. 1 | 6,659 | 4,148 | 839 | 6,461 |
| No. 2 | 7,621 | 7,492 | 1,506 | 7,687 |
| No. 3 | 7,459 | 4,800 | 1,351 | 7,301 |
| No. 4 | 6,794 | 4,152 | 658 | 6,594 |
| Total S.H.P., all shafts | 28,533 | 20,592 | 4,354 | 28,043 |
| I.H.P., all auxiliaries | 786 | 773 | 460 | ..... |
| horsepower, all machinery | 29,319 | 21,365 | 4,814 | ..... |

# THE KINGSBURY THRUST BEARING.

By W. W. Smith, Lieutenant, U. S. Navy, Member.

An interesting development in engineering is a thrust bearing invented by Mr. Albert Kingsbury, of Pittsburgh. In this bearing the load is uniformly distributed over the entire surface, and the slippers are free to adjust themselves at slight angles to the collar so as to glide or skim over a film of oil which adheres to and moves with the collar, the action of a slipper being much the same as that of a single hydroplane. As compared with the usual types, the results which have been obtained with this bearing are remarkable and of great importance. In high-speed bearings a pressure of 500 pounds per square inch is being carried, and in low-speed, 900 pounds. Although these pressures are unusually high, they are still far from the safe maximum pressure which can be carried, the factor of safety being ten or more. As noted below, a test on a turbine thrust bearing showed that a pressure of about 7,000 pounds per square inch could be carried without the breaking down of the oil film.

The bearings, both during shop tests and in actual service, have proved entirely satisfactory. The frictional losses, and consequently the temperature rise, under usual conditions are small, the coefficient of friction being from .0008 to .003, depending on the size and speed of the bearing, the unit pressure and the character of the oil. The wear has been practically *nil* because the surfaces are entirely separated by the oil film. No repairs, other than periodic examinations, have been required.

Kingsbury thrust bearings were fitted on the turbines of the U. S. S. *Neptune*, and it was the writer's experience that they gave most excellent results as regards both operation and repairs. In regard to the latter, it may be noted that,

because of the self-alignment of the bearing, any mechanic of fair ability can overhaul it without difficulty and with the assurance that it will run properly when assembled, which is, of course, important.

### THE "NEPTUNE'S" THRUST BEARING.

The following is a description of the *Neptune's* thrust bearing which is illustrated in Fig. 1.

The bearing consists of a body secured to the turbine casing, in which are mounted in spherical seats two rings which carry, also in spherical seats, the slippers which bear against opposite sides of a single thrust collar which is held on the shaft by a key and a nut. Each of the slippers, which are of steel with the rubbing surfaces lined with white metal, fits into a recess in the slipper ring which holds it in the correct position, preventing rotation around its longitudinal axis, and is held in its spherical seat by a helical spring held in compression between the slipper ring and the head of its retaining bolt which passes through a hole in the ring.

Each of the steel slipper rings, which are carried in the spherical seats of the body, is held against rotation and supported by a bolt which passes through a short slot with the spherical lower surface of its head resting on a thick washer on the supporting yoke; this yoke is carried by a transverse bolt passing through lugs on the body.

The cast-steel body consists of a forward part which is screwed into an after part on a diameter exceeding those of the thrust collar and slipper rings, which are contained in the latter. The bearing clearance or "float" is adjusted by screwing the forward part in or out; and, as the after part is divided on top by a longitudinal slot through the threaded portion, the two parts can be clamped in the required position by the transverse bolt which passes through lugs on each side of the slot. The forward part of the body extends through the casing to which its flange is bolted. Under the flange there are brass liners, made in halves, by means of which the bearing can be

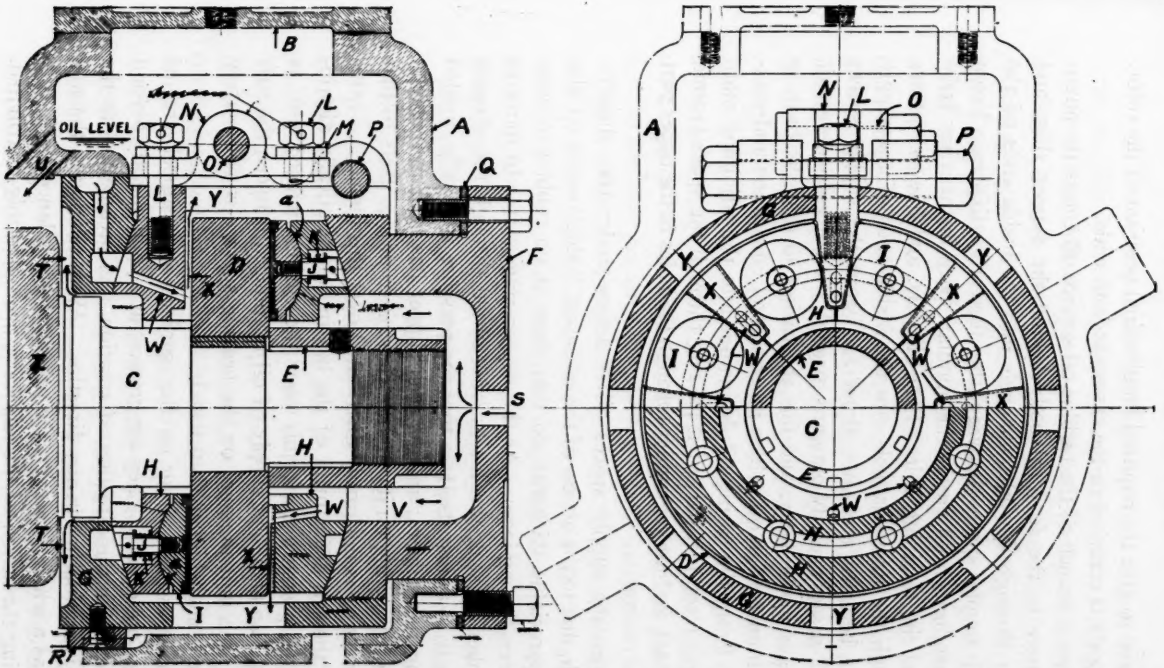



FIG. 1.—KINGSBURY THRUST BEARING OF TURBINES OF U. S. S. "NEPTUNE."—DIAMETER, 13½ INCHES; THRUST, 45,000 POUNDS.

A, casing. B, cover. C, shaft. D, thrust collar. E, collar nut. F, forward body. G, after body. H, slipper ring. I, slipper. J, Retaining bolt. K, retaining spring. L, supporting bolt. M, supporting washer. N, supporting yoke. O, yoke bolt. P, locking bolt. Q, adjusting liners. R, centering block. S, oil inlet. T, oil outlet. U, oil outlet. V, oil-inlet space. W, oil-inlet channel. X, oil-supply space. Y, oil-exit passage. Z, main bearing. *a*, spherical seat of slipper.

adjusted to give the required longitudinal position of the rotor. The body is centered in the casing at both ends.

From a branch of the turbine oil service oil enters the annular space in the forward body behind the slipper ring, and passes through the inlet channels into the inside ends of the radial supply spaces between the forward slippers. From another branch oil enters the annular space in the after body behind the retaining bolts of the slippers, and likewise passes through the inlet channels into the inside ends of the supply spaces between the after slippers. From the supply spaces the oil passes radially outward through the corresponding exit passages in the body, and fills the casing so that the bearing runs immersed in oil, which insures good lubrication and cooling. From the casing oil drains out into the adjoining main bearing reservoir, part of it passing through the space around the shaft, and the remainder through the holes in the upper part of the casing.

From the supply spaces oil is drawn under the slippers, which, due to the action of the lubricant, " ride" on an oil film of appreciable thickness, so that there is no rubbing contact between the surfaces. As the slipper ring is free to move in its spherical seat, the forces transmitted through the slippers cause it to adjust itself so that the pressure is equally divided among the slippers; and, as each slipper has similar freedom, the forces acting on its surface cause it to adjust itself so that the pressure is uniformly distributed over the surface. Therefore, the self-adjustment of the bearing automatically distributes the pressure uniformly over the entire surface; and, as there is an abundant supply of oil introduced between the surfaces, the unit pressure on the bearing surface is with safety carried much higher than usual.

The pressures acting on the surface of a slipper are, of course, not equal. They are greatest at a point slightly beyond the center in the direction of rotation, and become less as the edge is approached in any direction. With relation to the surface as a whole, however, the pressures are balanced and, considering the action of the lubricant, are uniformly distributed.

The oil drawn under the leading edge of a slipper escapes from the three remaining sides, the major part of it passing out under the following edge. Therefore, the thickness of the film is greatest under the leading edge and least under the following edge. This difference of thickness of the film causes the slipper to stand at a slight angle to the surface of the collar, and, so to speak, it "rides" on the film in a position which is required for the proper action of the lubricant. This is similar to the action in a cylindrical bearing, where the greater thickness of the film on the leading side causes the journal to run slightly eccentric to the bearing. However, due to the journal and bearing having approximately the same curvature, the action of the lubricant is not as perfect as in the case of the slippers with plane surfaces. In the usual types of thrust bearings, where the surfaces are held rigidly parallel, the action of the lubricant is still less perfect, which accounts for the low unit pressures to which they are limited.

The *Neptune's* bearing, which has been described, illustrates in a general way the mechanical features of all Kingsbury bearings. The details are varied somewhat to suit the conditions, but the principle of construction remains the same. The bearing shown in Figs. 2 and 3 is used to support the rotor of a large water-wheel generating unit. It will be noted that the slippers have plane surfaces which rest on spherical seats, which in this case are convex to the slipper. The slipper seats are provided with wedges and liners by means of which the vertical position of the rotor can be adjusted. Each slipper can be adjusted independently and removed radially as shown. The center of support for the slipper is slightly beyond the geometric center of the surface in the direction of rotation, and corresponds with the theoretical center of pressure. This construction gives greater freedom to the slipper and slightly reduces the coefficient of friction. Oil is admitted to the reservoir formed by the bearing casing, and passes out through an overflow pipe located at the oil level shown in the illustration. The bearing therefore runs in an oil bath and is subject to continuous circulation which removes the heat. In this type

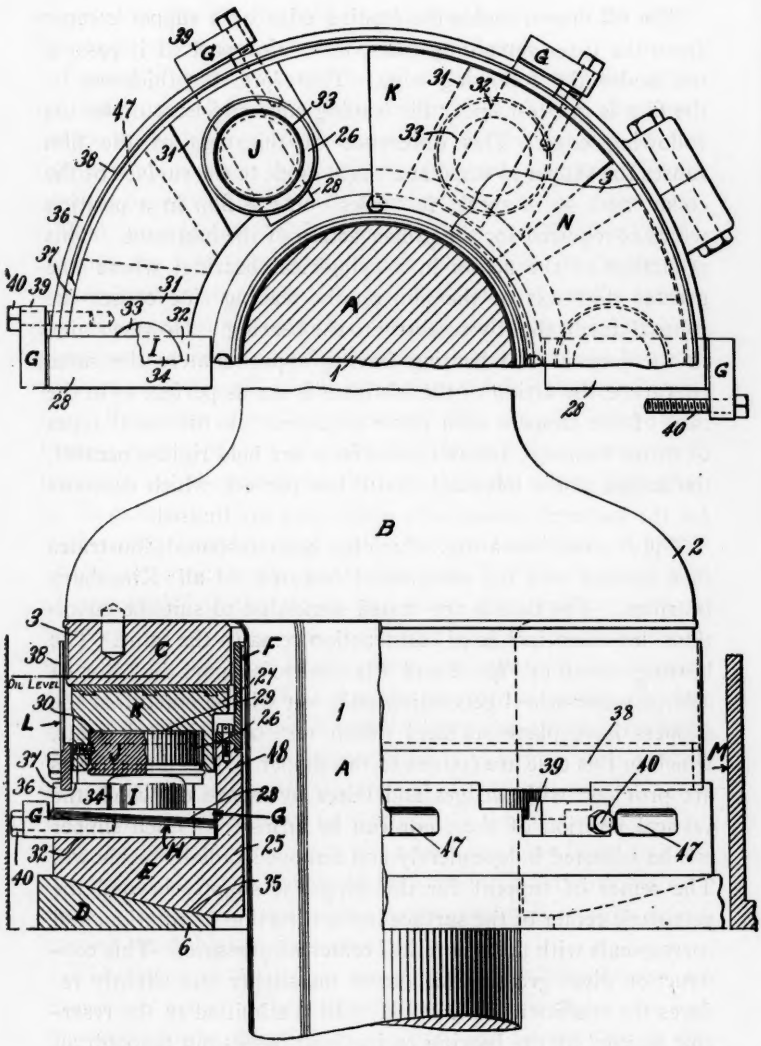



FIG. 2.—THRUST BEARING OF LARGE WATER-WHEEL GENERATING UNIT.

A, shaft. B, collar seat. C, thrust collar. D, bearing seat. E, slipper ring. F, inside oil-retaining ring. G, slipper-adjusting wedges. H, liners. I, slipper seat, lower. J, slipper seat, upper. K, slipper. L, outside oil-retaining ring. M, Casing. N, oil-supply space.

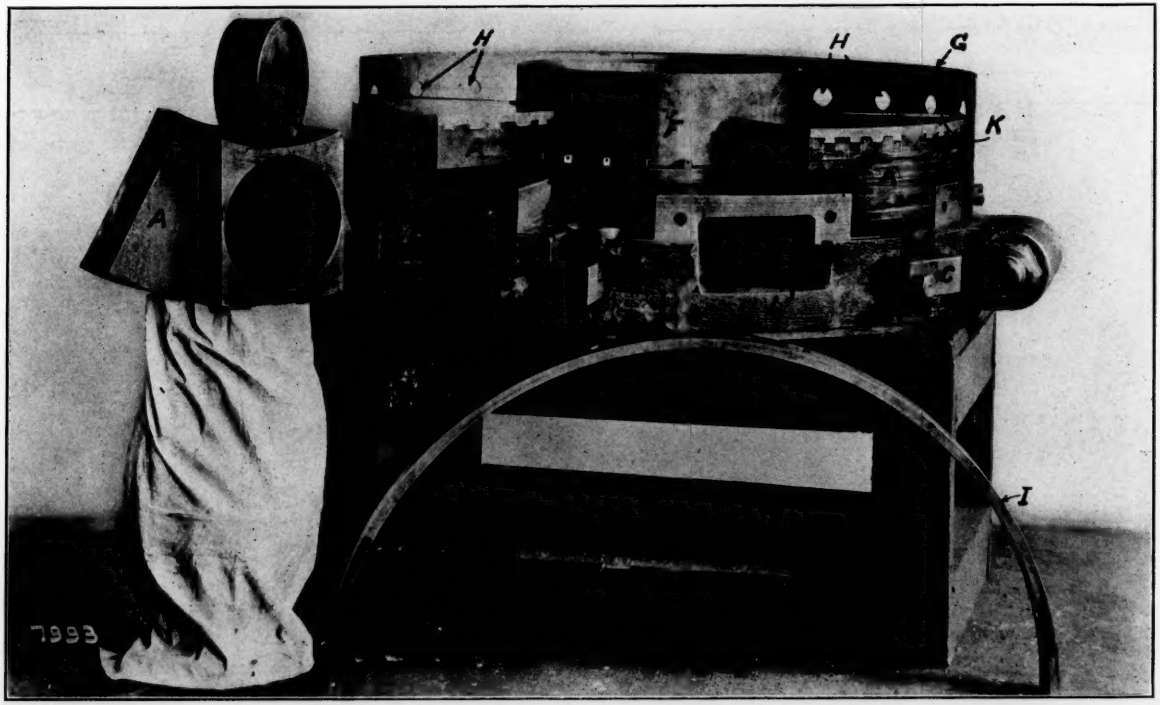



FIG. 3.—THRUST BEARING OF LARGE WATER-WHEEL GENERATING UNIT. DIAMETER, 48 INCHES; THRUST, 410,000 POUNDS.

A, Slipper. B, upper slipper seat. C, slipper-adjusting wedge. D, liners. E, lower-slipper seat. F, inside oil-retaining ring. G, outside oil-retaining ring. H, oil-exit holes. I, part of oil-retaining ring, outside. J, body. K, oil-supply space. L, oil-inlet passage.

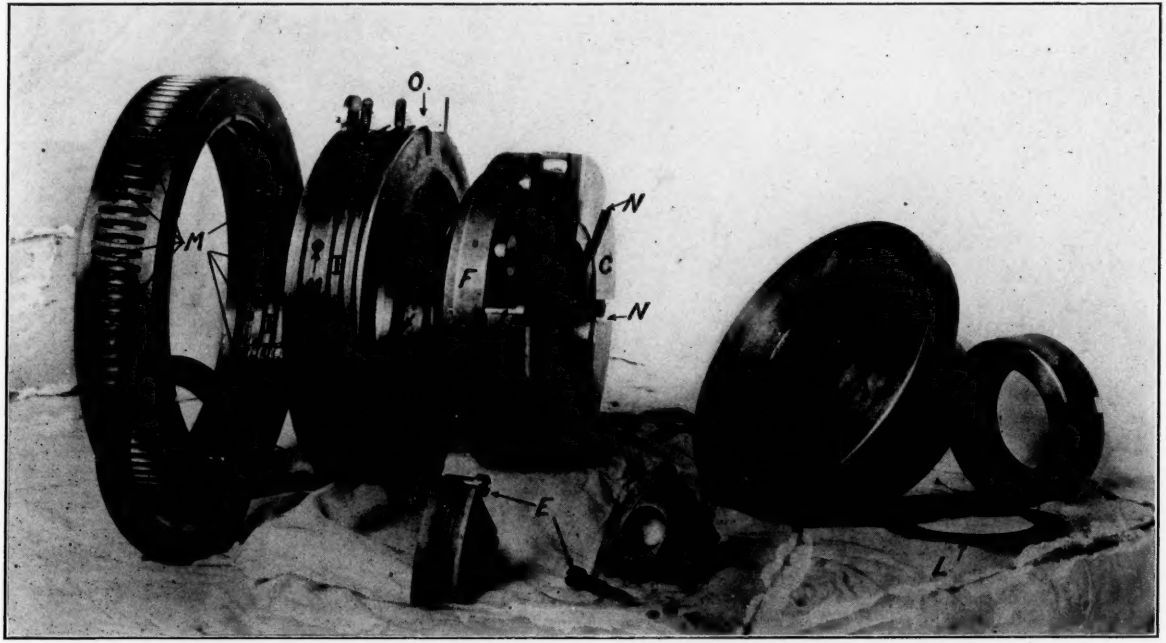



FIG. 4.—THRUST BEARING OF STEAM TURBINE. DIAMETER, 12 INCHES; THRUST, 40,000 POUNDS.

A, thrust collar. B, collar nut. C, slipper. D, slipper seat. E, retaining screw. F, slipper ring. G, bearing seat (spherical). H, dowel. I, adjusting threads. J, adjusting ring and worm rack. K, shaft-packing ring. L, collar-nut washer. M, oil holes. N, oil-supply space. O, locking device.

of bearing, where the thrust is always in the same direction, only one set of slippers is used. This type is also used in turbines and other horizontal machines where the thrust is always in the same direction. In marine turbines and other machines where the direction of the thrust changes, slippers must, of course, be provided on both sides of the collar as shown in the *Neptune's* bearing.

The single bearing illustrated in Fig. 4 is used in a steam turbine of a generator unit. This construction has been used in all turbine work of this class.

### TESTS TO DETERMINE PERMISSIBLE BEARING PRESSURES.

To determine the maximum pressures that could be carried on a Kingsbury thrust bearing, tests were made by The Westinghouse Machine Company on a bearing of the following dimensions: Outside diameter of collar, 4¾ inches. Inside diameter of collar, 2⅜ inches. Number of slippers, 10. Total area of slippers, 10.4 square inches (originally). The total pressure on the bearing was obtained from the pressure acting on the dummy piston of the turbine on which the bearing was tested. This pressure was regulated by throttling the steam in the equilibrium pipe, which caused the pressure to build up behind the dummy piston, producing a thrust which was carried by the bearing. The r.p.m. of the turbine were 3,440–3,490 during the tests, and the mean surface speed approximately 3,240 feet per minute.

The bearing was first loaded to 10,500 pounds, giving a unit pressure of 1,010 pounds per square inch, and ran satisfactorily. As the total thrust could not be increased sufficiently beyond this point, all but four of the slippers were removed, and, with a total load of 10,900 pounds and a unit pressure of 2,620 pounds, the bearing ran properly without heating. All but two slippers were then removed, and the bearing carried a total load of 11,300 pounds and a unit load of 5,420 pounds satisfactorily. Part of the surface of the two slippers was then removed, leaving 1.91 square inches, and the bearing carried a total load of 11,300 pounds and a unit load of 5,910

pounds without heating. The surface was then reduced to 1.18 square inches, and a total load of 11,700 pounds was carried without heating. During this test the babbitt lining of the slippers flowed, or was pressed out, on all sides, increasing the surface to 1.67 square inches, and giving a unit pressure of 7,000 pounds, based on the latter area. Another test with an area of 1.39 square inches and a total load of 9,800 pounds, gave a unit pressure of 7,050 pounds. In this case, also, the babbitt flowed slightly, but no heating occurred. Several other tests were made with the object of attaining a higher unit pressure, but in each case the lining flowed sufficiently to reduce it to about 7,000 pounds, which was the limit with this particular bearing. The unit pressure was, therefore, limited by the flowing pressure of the lining, and not by failure of lubrication. In all of the above tests the oil film was maintained, and, even under the enormous unit pressures which were carried, the lubrication was perfect. No cutting, or any other injury to the surfaces, other than the flowing of the lining noted above, occurred, and they remained in excellent condition throughout the tests. It is interesting to note that the babbitt flowed on the leading edge, or opposite to the direction of rotation, the same as on the other edges.

The slippers were lined with standard babbitt metal, the thickness being 1/16 inch. The thrust collar was made of steel, the surface being smooth, but not polished.

The foregoing shows the enormous unit pressures that can be carried, and makes it plain that working pressures of 500 to 1,000 pounds per square inch are low in comparison with the ultimate pressures that can safely be carried. This gives a large factor of safety, or overload capacity, which is a decided advantage.

The *Neptune's* thrust bearing carries a load of about 45,000 pounds, the unit pressure being 500 pounds and the surface speed 4,300 feet per minute. In the power plant of the Pittsburgh Railways Company there are three 5,000 kw. turbo-generator units in which Kingsbury bearings are in use. The bearings have been entirely satisfactory, and carry a unit load

of 500 pounds with a surface speed of 4,200 feet per minute. In general, turbine thrust bearings of this type are designed to carry about 500 pounds unit pressure, and usually have a surface speed of 3,500–4,500 feet per minute. A much higher unit pressure could be safely carried if desirable, but, in most cases, a larger bearing suits the general design better, and, of course, there is no objection to it.

The large bearing illustrated in Figs. 2 and 3 supports the rotor of a 17,500 H.P. vertical water-wheel generating unit at the power plant of the Pennsylvania Water & Power Company, Holtwood (McCall's Ferry), Pa. The following are data of its construction and a test under service conditions:

| | |
|---|---|
| Outside diameter of collar, inches | 48 |
| Total load on bearing, pounds | 410,000 |
| Area of slippers, square inches | 1,160 |
| Unit pressure per square inch, pounds | 350 |
| Revolutions per minute | 94 |
| Mean surface speed, feet per minute | 900 |
| Oil flow through bearing per minute, gallons | 15 |
| Kind of oil | Light mineral. |
| Mean temperature rise of oil in bearing, degrees F. | 4½ |
| Frictional loss in bearing, H.P. | 9 |
| Coefficient of friction | .0008 |

A test was made at a speed of 10 r.p.m., the surface speed being 95 feet per minute, and it was found that, at this low speed, the oil film was maintained, and that the lubrication was entirely satisfactory. The friction was much less than at full speed,—so small that no definite data as to its value could be obtained. This shows the great tenacity of the oil film, the large range of speed with perfect lubrication, and that, at normal speeds, the oil film is amply thick to prevent any contact whatever of the surfaces. It follows, of course, that no wear whatever occurs at normal speeds, and can only occur to a very minute extent at the instants of starting and stopping.

It is interesting to note that the coefficient of friction, shown by the above tests and verified by a number of others, is unusually low. It is lower than that of any other known type, including ball and roller bearings, for which the coefficients

are usually from .0015 to .003 under good conditions, and not excepting the hydraulically-supported step bearing, if we charge against the latter power required for the operation of the pumps.

The heaviest unit pressure carried in service by a bearing of this type is in the thrust bearing of a plate-glass grinding machine in the works of the Pressed Prism Plate Glass Co., Morgantown, W. Va. The data for this bearing are as follows:

| | |
|---|---|
| Total load, pounds | 160,000 |
| Unit pressure, pounds per square inch | 920 |
| Outside diameter of collar, inches | 18 |
| Mean surface speed, feet per minute | 125 |

A large bearing (36 inches in diameter) has been in operation on a steel rolling mill in the works of the Jones & Laughlin Steel Co., Pittsburgh, Pa., since April, 1911.

It is of interest and importance to note that, in all of the applications of this bearing, the operation has proved entirely satisfactory, none of the bearings having given trouble or required repairs.

In this connection it is also interesting to note that, in a large turbo-generator unit fitted with the Kingsbury bearing, the engineer accidentally permitted the oil to run out of the system, with the natural result that the main bearings burned out, causing considerable damage to the blading. which closed the steam passages and badly unbalanced the turbine. Although the load on the thrust bearing was greatly increased due to the excessive thrust, it carried the load without burning out, or, in fact, any injury whatever; and, by keeping the rotor in correct position endwise, prevented serious injury to the machine. The oil standing in the thrust-bearing casing afforded lubrication after the supply had ceased. Not being injured, the bearing, of course, required no repairs, and, when put into service again, ran perfectly.

In ruggedness and general reliability it is claimed that this bearing is superior to other types; and, in the opinion of the writer, this is true, as indicated by the above accident, the large overload capacity and the substantial construction.

This bearing also has a quality that is most important for marine turbine work, where it is desirable to reduce to a minimum the fore-and-aft movement of the rotor due to reversing. While this is most important in Parsons turbines it is also important in other types, because it permits smaller end clearances. This quality is the fine bearing clearance which is permissible with perfect lubrication, .002 inch being quite ample. Also, as no wear occurs, this adjustment does not change, and therefore less tinkering with the thrust bearing and end clearances is required, which, I believe, will be fully appreciated by operating engineers.

## FAULTS OF COLLAR THRUST BEARINGS.

As stated above, the lubrication of an ordinary collar thrust bearing is, because of its principle of construction, imperfect. That is, the surfaces are not entirely separated by the oil film, with the result that they rub, wear and produce friction. The wear, being well known, requires no further comment. To remove the considerable heat generated by friction water is circulated through the main thrust collars, and, as they usually remain cool, but little thought is given to the frictional losses which occur. The loss, however, is considerable and, obviously, it would be desirable to reduce it.

No direct observation as to the frictional loss in an ordinary thrust block has been made; but, from experimental data, such as Tower's well known tests, it appears probable that the coefficient of friction is not less than .02 when the bearing is in the best condition, and much higher than this when in bad condition. Because of the difficulty of fitting a thrust bearing properly, as pointed out below, it is believed that, in service, the coefficient is considerably greater. However, for comparison, it is desired to assume a good performance for this type, and the above value is taken; this assumption being necessary because of the lack of more precise data.

In a naval vessel, the power loss and coefficient of friction for the main thrust bearing could be easily determined by taking the temperatures of the cooling water before and after

passing through the shoes, and measuring the quantity of flow. As this information would be very valuable, it is hoped that some one will make the test and publish the results in the JOURNAL.

In a 30,000-H.P. battleship, the horsepower of the main engines required to overcome the frictional resistance in an ordinary thrust bearing would be 146 for one shaft and 292 for both, coefficient of friction being .02. Thus 292 horsepower, with the corresponding expenditure of fuel, is being constantly developed to produce useless work.

The coefficient of friction for the Kingsbury thrust bearing has been found by a large number of tests to vary from .0008 to .003, as noted above. A conservative value of the coefficient for a marine thrust bearing would be .002, and, for purposes of comparison, this value is assumed. Thus in the same vessel the frictional horsepower would be 14.6 for one shaft, and 29.2 for both. This would be a saving of 262.8 horsepower, or 90 per cent., over the ordinary type of thrust bearing. It would seem worth while to make such a large saving as 90 per cent. In smaller bearings the frictional loss is actually less, but proportionally the same; and hence it is of equal importance.

In addition to the above, it is difficult to construct or to adjust an ordinary thrust bearing so that the load will be divided equally among the collars. In this respect, as is known, an old bearing is usually better than a new one, because it has ground down the high places. These characteristics make it unsafe to carry high unit pressures on ordinary thrust bearings, and in safe practice, it is usually limited to 40–60 pounds. Because of the difficulty of fitting these bearings exactly right their action is erratic, and it is by no means a certainty that they will run without heating. The use of cooling water permits considerable heating without danger, and, so to speak, it covers a multitude of sins. However, even with water cooling, it is impossible to predict with certainty the action of the bearing and to provide a definite factor of safety. This can be and is done with the Kingsbury bearing. By reason

of the much higher unit pressures carried the Kingsbury bearing is smaller and lighter. Perhaps the most important advantage is the assurance that it will run properly after being overhauled by a mechanic of ordinary ability.

It is of interest to note that the entire thrust of 140,000 pounds on one of the shafts of the *North Dakota* could be taken on a single collar 29 inches in diameter with a unit pressure of 325 pounds; which, as seen from the above, is very moderate for a bearing of this type. In fact, it is probable that pressures higher than this would be carried in marine thrust bearings, because the reduction in size and weight would be advantageous.

As previously noted, the Kingsbury thrust bearing has been applied to steam turbines, both land and marine, vertical water-wheel generating units, steel rolling mills, plate-glass grinding machines, and vertical motors. Its application is much wider than this, and it can be used in nearly all machines, either horizontal or vertical, where other types are used. In certain cases, however, as, for example, very small bearings, its use would not be desirable. In marine work, it would appear that it could be applied with considerable advantage to main turbines, main or propeller-shaft thrust bearings, vertical forced-draft blowers, and vertical rotary pumps.

# COMPARATIVE TESTS OF THREE TYPES OF TURBINE-DRIVEN FORCED-DRAFT FANS.

By W. J. A. London.

On account of the increasing use of turbine-driven forced-draft fans in the Navy, the following tests were carried out at the works of the Terry Steam Turbine Co., Hartford, Conn., to determine the relative efficiency of three distinctive types of fans when combined with a standard Terry vertical turbine. The tests were witnessed by Capt. R. B. Higgins, U. S. N., and by Lieut. Comdr. John Halligan, Jr., U. S. N., representing the Navy Department.

The turbine was the maker's standard type "BV," being a duplicate of over 50 machines now in use in the Navy. The normal rated capacity was 60 H.P., with a throttle pressure of 250 pounds and a terminal pressure of 10 pounds gage, at a speed of 1,400 r.p.m.

The turbine was equipped with four diverging nozzles, one nozzle being fitted with hand valve. The thrust is taken by a Hess-Bright ball bearing with spherical seat. Glands of the standard floating ball-ring bushing type were used. Forced circulation of oil to thrust and all bearings is obtained by means of a self-contained gear pump.

The first fan tested (Plate I and Fig. 1) will be referred to as the Type A fan.

The rated capacity was 23,000 cubic feet of free air against 5 inches static pressure by water gage, speed of 1,400 r.p.m.

The second fan tested (Fig. 2) will be referred to as the Type B fan. Its rated capacity was 23,000 cubic feet of free air against 5 inches static pressure by water gage, at a speed of 2,200 r.p.m.

The third fan tested (Fig. 3) will be referred to as the Type C fan. This fan was rated at 23,000 cubic feet of free

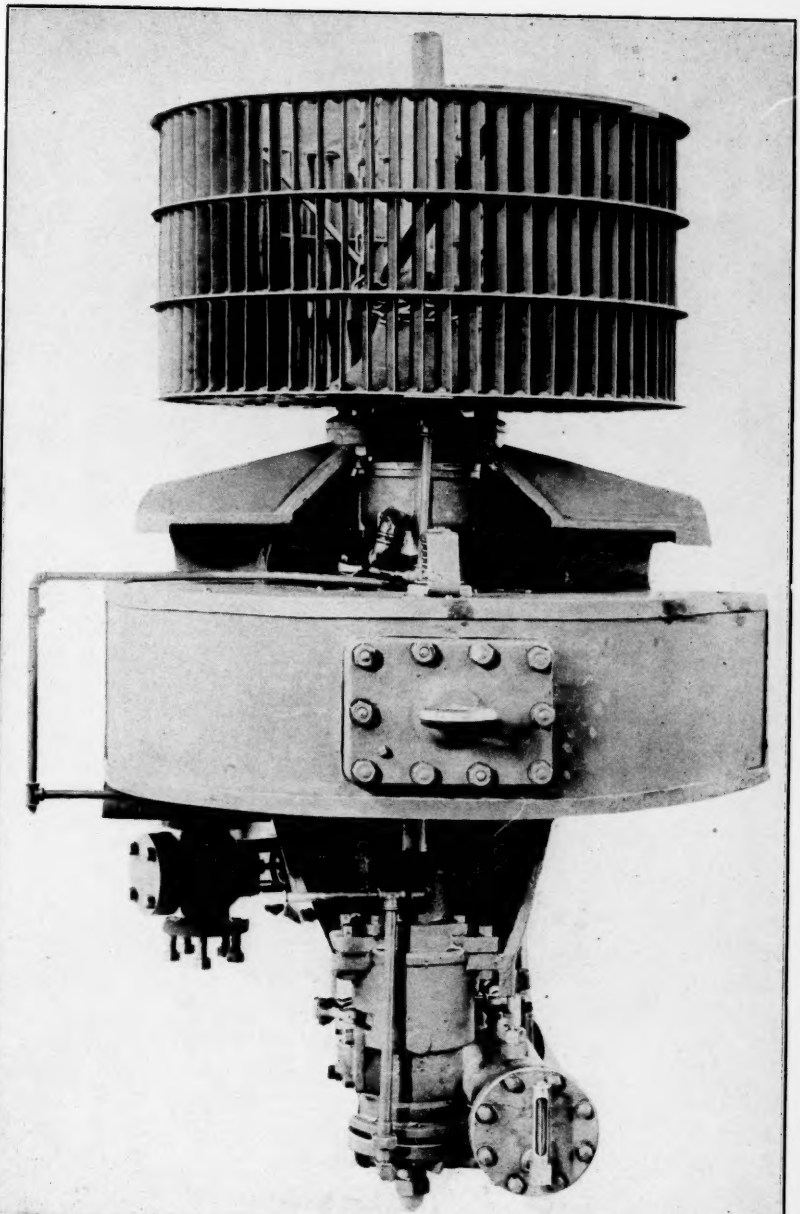


PLATE I.—60-H P. VERTICAL TURBINE BLOWER. TERRY TURBINE. TYPE A FAN.

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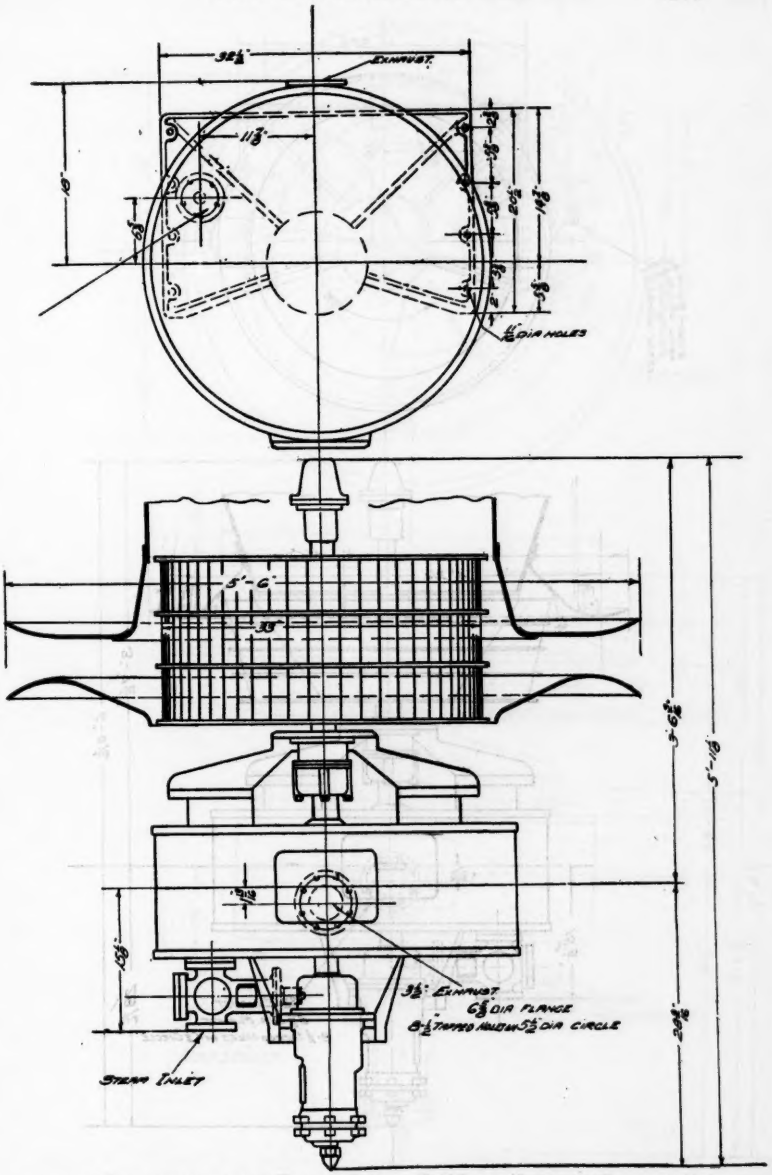



FIG. 1.—OUTLINE, TYPE BV TERRY STEAM TURBINE WITH 33-INCH TYPE A FAN.

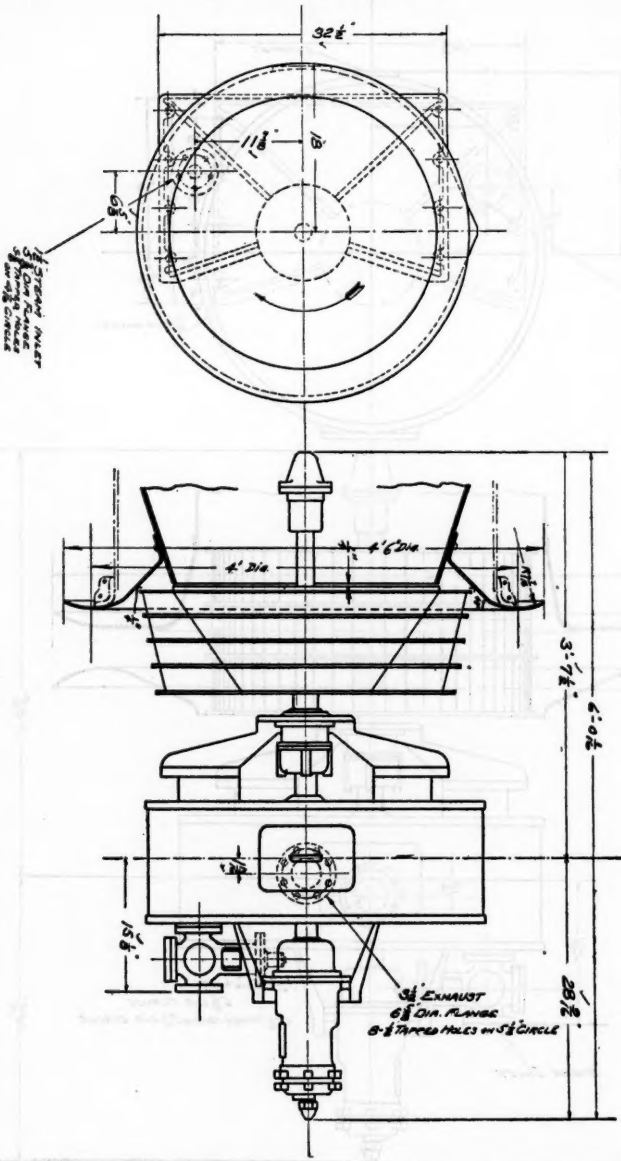



FIG. 2.—OUTLINE, TYPE BV TERRY STEAM TURBINE WITH 30-INCH TYPE B FAN.

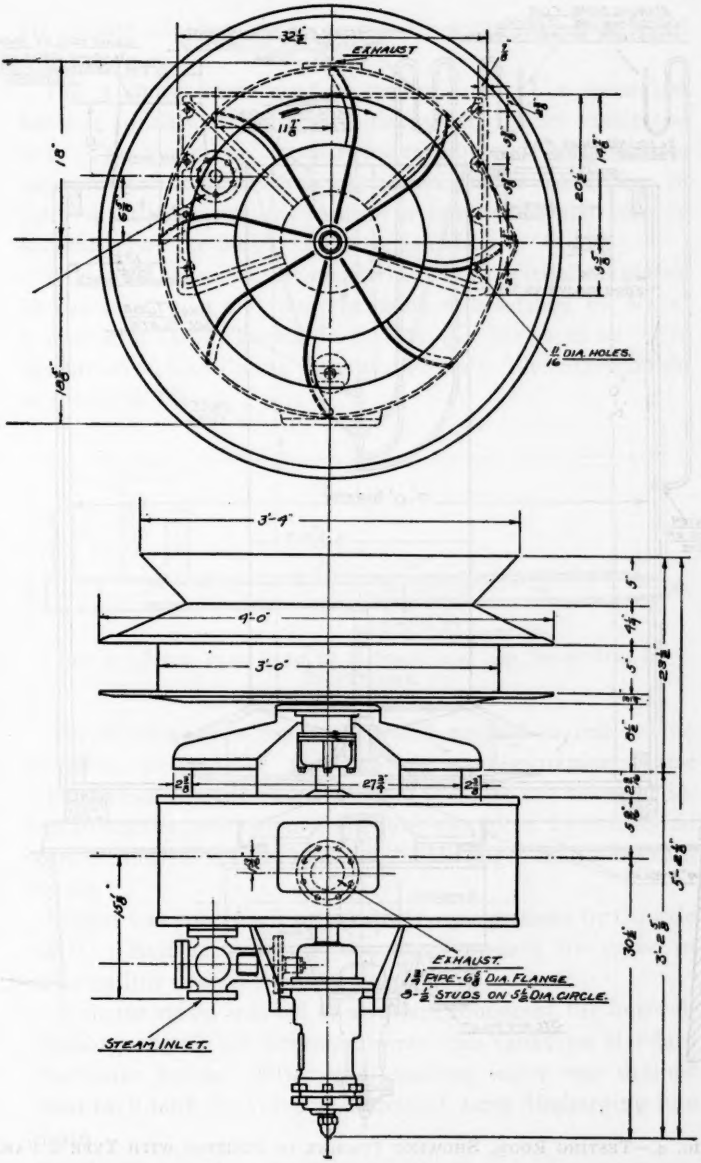



FIG. 3.—OUTLINE, TYPE BV TERRY STEAM TURBINE WITH 36-INCH TYPE C FAN.

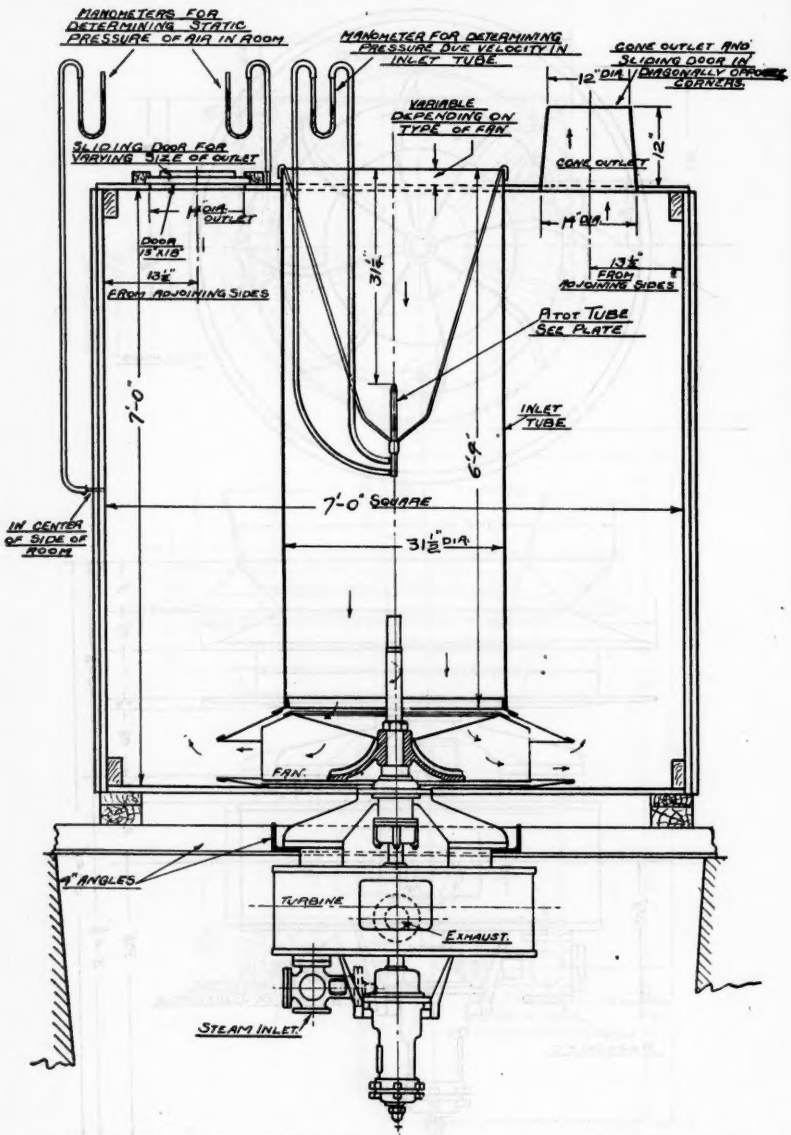



FIG. 4.—TESTING ROOM, SHOWING TURBINE IN POSITION WITH TYPE C FAN.

air against 5 inches static pressure by water gage, at a speed of 1,400 r.p.m.

Fig. 4 shows room used in testing fans. The room was built of $\frac{7}{8}$-inch matched boards, being well braced inside and out. The angle-iron supports for turbine were also used to support room. The drawing shows the Type C fan in position, it being possible to raise or lower the inlet tube to accommodate the height of the various fans.

The pitot tube used for measuring air velocities was placed in the inlet tube and was the same as described by Naval Constructor D. W. Taylor, U. S. Navy, in his paper on "Experiments on Ventilating Fans and Pipes." The construction is shown in Fig. 5.

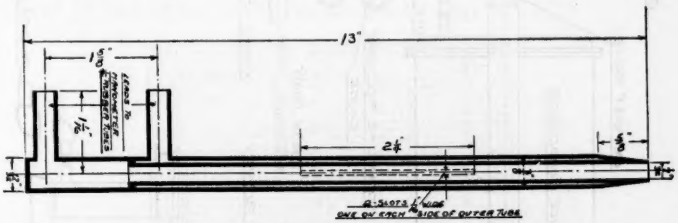



FIG. 5.—PITOT TUBE USED IN DETERMINING AIR VELOCITIES IN INLET TUBE.

Fig. 6 shows pipe connections and method of test. Line pressure, temperature and calorimeter measurements were taken before throttle valve. Steam pressure was taken at inlets to nozzles, and exhaust pressure was taken by compound gage connected to turbine casing. All gages used were calibrated.

Speed was controlled by varying nozzle pressure by throttle valve. Exhaust pressure was kept constant by valve in exhaust line near turbine.

Exhaust steam was led to a surface condenser, the hot-well discharge being led alternately into two tanks on standard Fairbanks scales. After each reading water was drained from each tank by valve at bottom of same discharging into drain.

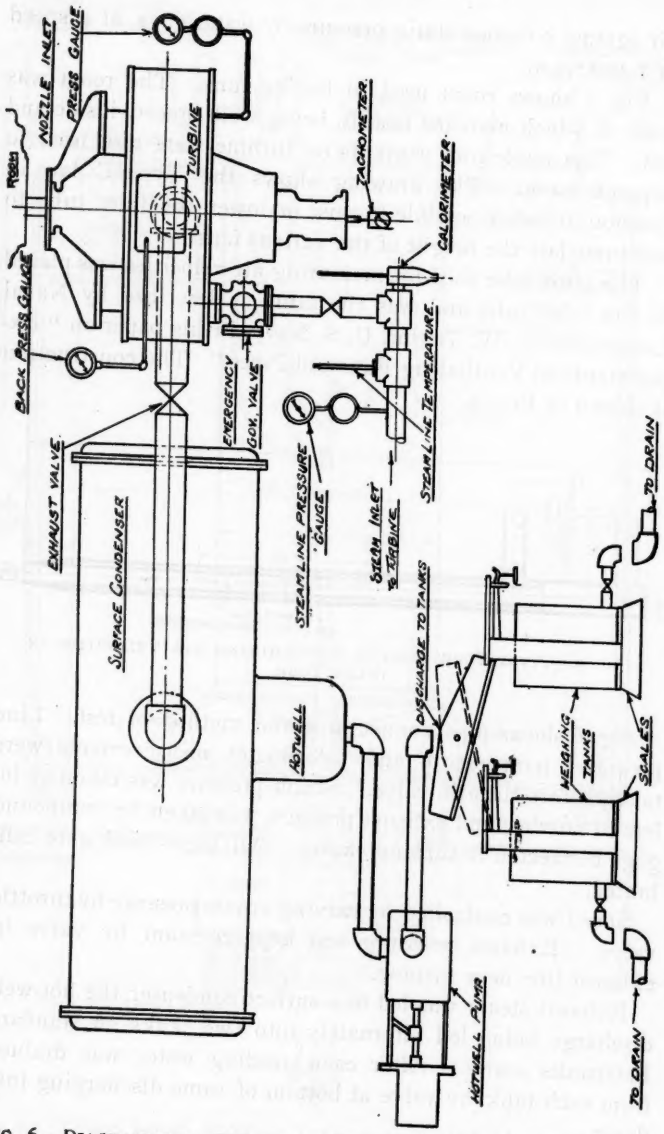



FIG. 6.—DIAGRAMMATIC OUTLINE OF PIPING AND METHOD OF TESTING.

The method of making test was as follows: Turbine was gradually brought up to normal rated speed by throttle, the operator on top of room keeping air pressure constant at five inches. This was done by varying outlet opening by means of a sliding door (See Fig. 4). When desired speed was reached nozzle pressure was noted and kept constant during run. Speed was read by a sensitive hand tachometer applied to lower end of turbine shaft. Static pressure was determined by two manometers, one connected to top and the other to side of room. Readings were then taken of steam pressures and temperatures, of air pressure and temperature, and pressure due to velocity of air entering fan, which latter was determined by direct-reading manometer connected to double pitot tube placed in inlet tube to fan, as shown in Fig. 4.

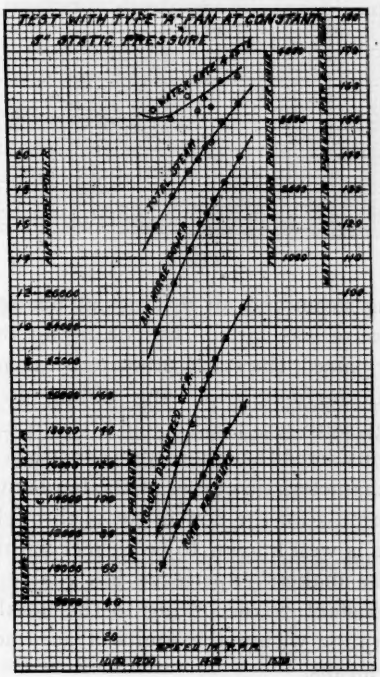



FIG. 7.

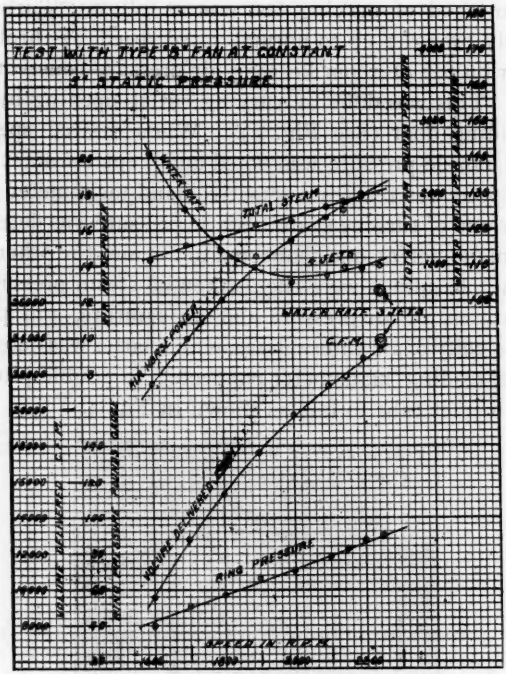



FIG. 8.

After taking steam and air readings at one speed the nozzle inlet pressure was varied, thus changing the speed and amount of air passed, the static pressure being maintained at five inches throughout by closing sliding door. Air and steam readings were again taken, and this procedure repeated until it was considered that there were enough points obtained to show the characteristic curves of the fan being tested.

Napier's formula for the flow of steam through a correctly-formed orifice is well known to be correct within a very few per cent. The makers' experience has enabled them so to modify the constant 70, according to the nozzle used, as to bring this formula within the limits of accuracy obtainable by any other means.

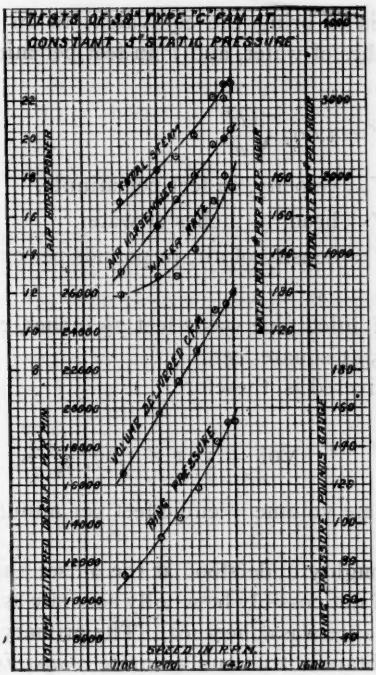

RESULTS OF 36" THREE G. FAN AT CONSTANT 3" STATIC PRESSURE
AIR DISCHARGE
TOTAL STEAM PRESSURE
VOLUME OF AIR PER MIN.
SPEED IN R.P.M.

FIG. 9.

Napier's formula reads, $\frac{PA}{70} \times 3{,}600$ equals pounds steam per hour.

For the jets in question the constant 72, instead of 70, had been previously determined. The accuracy of this method is well demonstrated by a comparison of the calculated quantities and the check figures taken at various points by actual measurements of the hot-well discharge.

In calculating velocity of air in inlet pipe, the following method was used, depending on the fundamental formula for velocity of fluids, namely:

$$V = \sqrt{2gH};$$

where $V$ = velocity, in feet per second,

$H$ = head, in feet;

modifying this to bring the equivalent to an air head, expressed in inches of water, and $V$ in feet per minute,

$$V = 1{,}097\sqrt{\frac{H}{d}},$$

where $V$ = velocity, in feet per minute,
$H$ = head, in inches of water,
$d$ = density of air, in pounds per cubic foot.

Diameter of inlet pipe was 2 feet $7\frac{1}{2}$ inches, corresponding to an area of 5.41 square feet.

Volume delivered is, therefore,

$$\text{Vol.} = 5.41 \times 1{,}097\sqrt{\frac{H}{d}},$$

where volume = cubic feet per minute.

It was thoroughly understood at the outset that the results obtained could not possibly be wholly accurate as far as the absolute performance was concerned. The primary object of the tests was to obtain a true and accurate *comparison* only, and special attention was paid to obtain this end. In order, how-

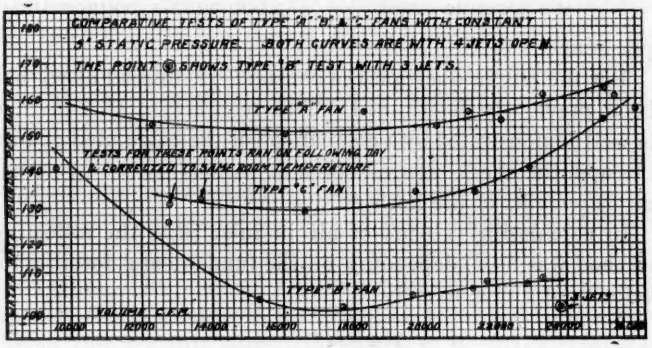

COMPARATIVE TESTS OF TYPE 'A' 'B' & 'C' FANS WITH CONSTANT
3" STATIC PRESSURE. BOTH CURVES ARE WITH 4 JETS OPEN.
THE POINT ◎ SHOWS TYPE 'B' TEST WITH 3 JETS.
TYPE "A" FAN
TESTS FOR THESE POINTS RAN ON FOLLOWING DAY
& CORRECTED TO SAME ROOM TEMPERATURE
TYPE "C" FAN
TYPE "B" FAN
3 JETS
VOLUME C.F.M.

FIG. 10.

ever, to make these tests approximate actual conditions a correcting factor was assumed throughout, which brought the resultant figures as near as possible to the guaranteed figures of the makers. This correction figure for volumes obtained was taken as 0.98.

Results of tests are shown in Tables 1, 2 and 3, and Figs. 7, 8 and 9.

It being understood that the point of primary importance is the absolute water rate in pounds per air H.P., a final curve, Fig 10, has been plotted showing the results obtained by all three fans on a basis of total air delivered at a constant pressure of 5 inches W. G., together with the corresponding water rates in each case.

TABLE 1.—RESULTS OF TEST ON TYPE A FAN, SEPT. 12, 1911.

FOUR NOZZLES OPEN.

| Test letter. | Revolutions per minute. | Air density, lbs. per cubic foot. | Velocity head, ins. of water. | Volume delivered, C. F. M. | Static pressure in room, ins. of water. | Air horsepower. | Nozzle inlet pressure, lbs. gage. | Nozzle inlet pressure, lbs. absolute. | Total water by Napier's formula, lbs. per hour. | Total water, actual, corresponding to dry steam, lbs. per hour. | Water rate, lbs. per air H.P. hour, calculated. | Water rate, lbs. per air H.P., hour, actual. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| A | 1,400 | .0726 | .97 | 21,250 | 5 | 16.72 | 121.7 | 136.4 | 2,625 | 2,638 | 156.8 | 157.6 |
| B | 1,450 | .0725 | 1.175 | 23,400 | 5 | 18.41 | 140 | 154.7 | 2,980 | ... | 161.7 | ... |
| C | 1,500 | .0724 | 1.35 | 25,100 | 5.05 | 19.96 | 154.5 | 169.2 | 3,257 | ... | 163 | ... |
| D | 1,350 | .0724 | .725 | 18,400 | 5 | 14.48 | 103.5 | 118.2 | 2,275 | ... | 157 | ... |
| E | 1,300 | .0723 | .553 | 16,080 | 5 | 12.65 | 84.3 | 99.0 | 1,905 | 1,906 | 150.7 | 150.7 |
| F | 1,250 | .072 | .325 | 12,350 | 5 | 9.73 | 63 | 77.7 | 1,494 | ... | 153.5 | ... |
| G | 1,420 | .072 | 1.05 | 22,200 | 5 | 17.46 | 125 | 139.7 | 2,690 | ... | 154 | ... |
| H | 1,380 | .07185 | .885 | 20,400 | 5 | 16.05 | 112.75 | 127.45 | 2,452 | ... | 152.7 | ... |

TABLE 2.—RESULTS OF TEST ON TYPE B FAN, SEPT. 13, 1911.

| Test letter. | Revolutions per minute. | Air density, lbs. per cubic foot. | Velocity head, ins. of water. | Volume delivered, C.F.M. | Static pressure in room, ins. of water. | Air horsepower. | Nozzle inlet pressure, lbs. gage. | Nozzle inlet pressure, lbs. absolute. | Total water by Napier's formula, lbs. per hour. | Total water, actual, corresponding to dry steam, lbs. per hour. | Water rate, lbs. per air H.P. hour, calculated. | Water rate, lbs. per air H.P. hour, actual |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Four nozzles open. | | | | | | |
| A | 2,250 | .07465 | 1.20 | 23,320 | 5 | 18.36 | 90.7 | 105.4 | 2,028 | 2,043.6 | 110.4 | 111.2 |
| B | 2,200 | .0743 | 1.15 | 22,900 | 5 | 18 02 | 87.5 | 102.2 | 1,966 | ... | 109 | ... |
| C | 2,150 | .0742 | 1.0375 | 21,750 | 5 | 17.11 | 83 | 97.7 | 1,880 | ... | 109.8 | ... |
| D | 2,100 | .0743 | 1.00 | 21,320 | 5 | 16.78 | 79 | 93.7 | 1,803 | ... | 107.4 | ... |
| E | 2,000 | .0743 | .85 | 19,660 | 5 | 15.47 | 70.5 | 85.2 | 1,638 | ... | 105.8 | ... |
| F | 1,900 | .074 | .685 | 17,700 | 5 | 13.93 | 67 | 81.7 | 1,571 | ... | 112.7 | ... |
| G | 1,800 | .074 | .515 | 15,350 | 5 | 12.08 | 57 | 71.7 | 1,380 | ... | 114.2 | ... |
| H | 1,700 | .0739 | .354 | 12,720 | 5 | 10.01 | 51 | 65.7 | 1,264 | 1,248 | 126.3 | 124.7 |
| I | 1,600 | .0739 | .20 | 9,570 | 5 | 7.53 | 40.5 | 55.2 | 1,062 | ... | 141.1 | ... |
| | | | | | | Three nozzles open. | | | | | | |
| J | 2,250 | .0739 | 1.244 | 23,880 | 5 | 18.79 | 119.4 | 134.1 | 1,935 | 1,934 | 103 | 103 |

TABLE 3.—RESULTS OF TEST ON TYPE C FAN, SEPT. 15, 1911.

| Test letter. | Revolutions per minute. | Air density, pounds per cubic foot. | Velocity head, inches of water. | Volume delivered, C. F. M. | Static pressure in room, inches of water. | Air H.P. | Nozzle inlet pressure, pounds gage. | Nozzle inlet pressure, pounds, absolute. | Total water, Napier's formula, pounds per hour. | Total water, actual, corresponding to dry steam, lbs. per hour. | Water rate, pounds per A.H.P. hour, calculated. | Water rate, pounds per A.H.P. hr., actual. | Water rates correcting tests H. and I. to previous air temp. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| A | 1,350 | .0738 | 1.378 | 25,100 | 5 | 19.75 | 143.6 | 158.3 | 3,047 | 3,066 | 154.2 | 155.1 | ... |
| B | 1,400 | .07375 | 1.475 | 26,020 | 5 | 20.49 | 153 | 167.7 | 3,228 | ... | 157.5 | ... | ... |
| C | 1,300 | .07375 | 1.15 | 23,000 | 5 | 18.1 | 118.5 | 133.2 | 2,564 | ... | 141.6 | ... | ... |
| D | 1,250 | .0737 | 1.0 | 21,420 | 5 | 16.85 | 103.4 | 118.1 | 2,272 | 2,272 | 134.8 | 134.8 | ... |
| E | 1,200 | .0736 | .85 | 19,750 | 5 | 15.55 | 94 | 108.7 | 2,091 | ... | 134.5 | ... | ... |
| F | 1,100 | .07345 | .6 | 16,625 | 5 | 13.08 | 73.25 | 87.95 | 1,692 | ... | 129.3 | ... | ... |
| G | 1,380 | .07345 | 1.4 | 25,400 | 5 | 20.00 | 152.5 | 167.2 | 3,219 | ... | 160.8 | ... | ... |
| H | 1,050 | .0723 | .35 | 12,790 | 5 | 10.07 | 53 | 67.7 | 1,303 | ... | 129.5 | ... | 130.8 |
| I | 1,070 | .0723 | .4 | 13,660 | 5 | 10.75 | 58.5 | 73.2 | 1,408 | ... | 131 | ... | 132.3 |

# U. S. S. *FANNING.*

## CONTRACT TRIALS PERFORMANCE.

By Henderson B. Gregory, Associate.

The *Fanning,* Torpedo-Boat Destroyer No. 37, is one of the six vessels authorized by an Act of Congress approved June 24, 1910. The vessel was built under contract by the Newport News Shipbuilding and Dry Dock Company, of Newport News, Va., the price being $630,500.00, and the time of construction twenty-four months from date of contract, December 6, 1910.

She is a triple-screw Parsons turbine vessel, designed to develop 12,000 S.H.P. at about 775 revolutions per minute of the main turbines, the contract speed being 29.5 knots and the displacement about 742 tons.

The vessel and machinery are in all essentials similar to the *Roe* and *Terry,* built by the same company several years previous, and fully described in the February, 1911, number of the Journal of the American Society of Naval Engineers, which description also applies to this vessel.

### TRIALS.

The following trials were required:

(*a*) A progressive trial over a measured-mile course for standardizing the screws.

(*b*) A full-speed trial of four hours' duration in the open sea at the highest speed obtainable. The average speed to be at least 29.5 knots, with the average air pressure in the fire-rooms, not exceeding 5 inches of water and the steam pressure at the M.H.P. turbine not to exceed 240 pounds above the atmosphere.

(*c*) An endurance and fuel-oil and water-consumption trial of four hours' duration in the open sea, at as nearly as possible a uniform speed of 25 knots, the average not to fall below that figure.

(*d*) An endurance and fuel-oil and water-consumption trial at as nearly as possible an average of 16 knots.

The various trials were conducted on the 28th and 29th of last May. The standardization trial was run on the measured-mile course off the Delaware Breakwater, followed on the same

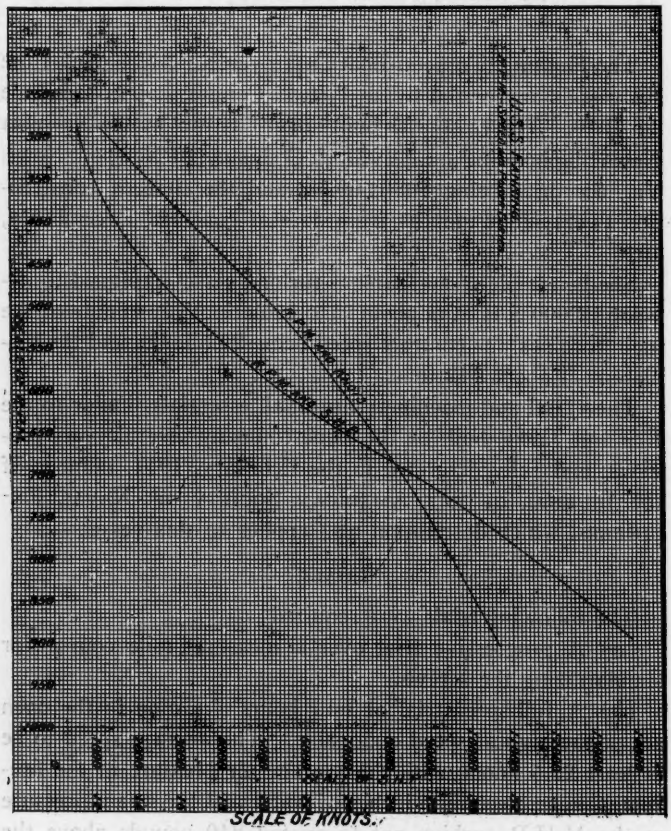



FIG. 1.

day by the 16-knot endurance trial, which was run in the lower part of Delaware Bay. The other trials were run in deep water in the open sea off Five Fathom Bank Lightship. The weather was favorable on all the trials.

## TABLE I.—STANDARDIZATION TRIAL DATA, U. S. S. "FANNING."

### Course off Delaware Breakwater. May 28, 1912.

| Number of run | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Time on course, min. sec. | 5 20 68 | 4 17.66 | 5 11.16 | 3 34.82 | 3 55.37 | 3 38.12 | 3 21.79 | 3 12.32 | 3 15.51 | 2 50.56 | 2 34.08 | 2 53.3 | 2 21.59 | 2 39.53 | 2 18.94 | 2 22.34 | 2 2.89 | 2 20.74 | 1 55.93 | 2 12.76 | 1 56.55 | 2 4.73 | 1 50.54 | 2 3.55 | 1 50.52 | 2 3.74 |
| Revs. to go 1 knot: | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Shaft No. 1 | 1,651.6 | 1,366.5 | 1,613.2 | 1,444.4 | 1,558.5 | 1,466.6 | 1,542.9 | 1,514.4 | 1,514.4 | 1,700.7 | 1,516.8 | 1,733.4 | 1,548.2 | 1,793 | 1,538 | 1,721 | 1,479.2 | 1,728.1 | 1,563.5 | 1,805.3 | 1,565.3 | 1,862.4 | 1,633.2 | 1,860.8 | 1,634.7 | 1,860.2 |
| No. 2 | 1,397.6 | 1,166.4 | 1,367.4 | 1,233.2 | 1,328.2 | 1,258.7 | 1,311 | 1,295.5 | 1,288.6 | 1,462.7 | 1,311.9 | 1,500.8 | 1,340 | 1,549.8 | 1,331.6 | 1,688.1 | 1,455.2 | 1,680.7 | 1,478.5 | 1,694.2 | 1,474.9 | 1,735.7 | 1,530 4 | 1,737.2 | 1,528 9 | 1,737.2 |
| No. 3 | 1,685.6 | 1,371.4 | 1,641.6 | 1,450.8 | 1,590.7 | 1,479.4 | 1,568.4 | 1,514.6 | 1,536.4 | 1,487.4 | 1,345 5 | 1,523.2 | 1,360.6 | 1,561.8 | 1,352.9 | 1,716.2 | 1,491.3 | 1,725.4 | 1,578.8 | 1,811.4 | 1,580.2 | 1,870 | 1,652.6 | 1,869.1 | 1,655.3 | 1,871.6 |
| Average | 1,578.3 | 1,301.4 | 1,540.7 | 1,376.1 | 1,492.5 | 1,401.6 | 1,474.1 | 1,441.5 | 1,446.5 | 1,550.2 | 1,391.4 | 1,585 8 | 1,416.3 | 1,634.9 | 1,474.2 | 1,708.5 | 1,475.2 | 1,711.4 | 1,520.3 | 1,770.3 | 1,540.1 | 1,822.7 | 1,605.4 | 1,822.4 | 1,606.3 | 1,823 |
| Revs. per minute | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Shaft No. 1 | 309.03 | 318.19 | 311.33 | 403.25 | 397.26 | 403.35 | 458.89 | 472.33 | 464.75 | 598.39 | 590.79 | 600.13 | 655.7 | 674.36 | 663.65 | 725.6 | 722.22 | 736.61 | 808.84 | 816.08 | 805.98 | 895.64 | 886.67 | 903.3 | 888.92 | 902 |
| No. 2 | 261.5 | 271.06 | 263.89 | 344.31 | 338.56 | 346.16 | 389.9 | 404.08 | 395.46 | 514.65 | 510.98 | 519.6 | 567 54 | 582.88 | 574.88 | 711.68 | 710.5 | 716.42 | 765.84 | 765.77 | 758.91 | 834.72 | 830.85 | 843.32 | 831.35 | 842.38 |
| No. 3 | 315.41 | 319.34 | 316.84 | 405.05 | 405.47 | 406.87 | 466.46 | 472.39 | 471.5 | 523.32 | 524.07 | 527.38 | 576.26 | 587.44 | 583.8 | 723.55 | 728.13 | 735.48 | 817.77 | 818.74 | 813.67 | 899.31 | 897.23 | 907.35 | 900.15 | 907.55 |
| Average | 295.31 | 302.86 | 297.35 | 384.2 | 380.43 | 385.46 | 438.42 | 449.6 | 443 9 | 545.45 | 541.95 | 549.04 | 599.83 | 614.88 | 607.34 | 720.28 | 720.28 | 729.5 | 797.82 | 800.2 | 792.85 | 876.56 | 871.58 | 884.66 | 873.47 | 883.98 |
| Speed, in knots | 11.225 | 13.972 | 11.57 | 16.76 | 15.296 | 16.505 | 17.838 | 18.718 | 18.413 | 21.108 | 23.363 | 20.773 | 25.426 | 22.563 | 25.910 | 25.292 | 29.294 | 25.579 | 31.053 | 27.116 | 30.888 | 28.862 | 32.568 | 29.139 | 32.573 | 29.092 |
| S.H.P., Shaft No. 1 | 243 | 250 | 252 | 518 | 511 | 518 | 786 | ... | ... | 1,965 | 1,940 | 1,957 | 2,637 | 2,712 | 2,685 | 3,349 | 2,922 | 2,945 | 3,585 | 3,593 | 3,587 | 4,199 | 4,199 | 4,213 | 4,210 | 4,251 |
| No. 2 | 125 | 129 | 126 | 262 | 266 | 272 | 408 | ... | ... | 1,090 | 1,107 | 1,249 | 1,540 | 1,595 | 1,600 | 3,252 | 3,297 | 3,291 | 4,046 | 3,990 | 3,919 | 4,648 | 4,726 | 4,757 | 4,709 | 4,711 |
| No. 3 | 385 | 289 | 272 | 578 | 579 | 494 | 855 | ... | ... | 1,021 | 1,047 | 1,017 | 1,426 | 1,468 | 1,431 | 3,065 | 3,067 | 3,081 | 3,892 | 3,897 | 3,951 | 4,473 | 4,570 | 4,621 | 4,584 | 4,622 |
| Total, all shafts | 753 | 668 | 650 | 1,358 | 1,356 | 1,284 | 2,049 | ... | ... | 4,076 | 4,094 | 4,223 | 5,603 | 5,775 | 5,716 | 9,666 | 9,286 | 9,317 | 11,523 | 11,480 | 11,457 | 13,320 | 13,495 | 13,591 | 13,503 | 13,584 |
| No. turbines used | Five. | | | | | | | | | Four. | | | | | | Three. | | | | | | | | | | |

On the standardization trial twenty-six runs, at various speeds, were made over the measured mile, the data from which were used in plotting the curves shown in Figure 1. The data are given in Table I.

From the official curves the following revolutions per minute of the main turbines were found to be necessary to attain the speeds stipulated for the other trials:

| Speed in Knots. | R.p.m. |
|---|---|
| 16 | 383 |
| 25 | 638 |
| 29.5 | 819 |

The full-speed and endurance trials were successfully run on the dates noted in Table II, which also gives a synopsis of the data obtained. The contract speed was comfortably exceeded on the full-speed trial without forcing, and all requirements were easily fulfilled on all the trials.

TABLE II.—TRIAL DATA, U. S. S. "FANNING."

| ..... | 4-hour full-power trial. | 4-hour 25-knot trial. | 4-hour 16-knot trial. |
|---|---|---|---|
| | 1912. | 1912. | 1912. |
| Date of trial | May 29. | May 29. | May 28. |
| Speed, in knots | 29.994 | 25.049 | 15.964 |
| Slip of propellers, per ct. of own speed, mean: | | | |
| Starboard propeller | 26.51 | 23.75 | 15.85 |
| Middle propeller | 21.38 | 12.42 | 2.77 |
| Port propeller | 27.45 | 16.51 | 17.28 |
| Draught, mean on trial, feet and inches | 8-4 | 8-4¼ | 8-4$\frac{11}{32}$ |
| Displacement corresponding, tons | 725 | 736 | 744.1 |
| Number of turbines used | 3 | 4 | 5 |
| boilers used | 4 | 4 | 4 |
| oil burners used (9 per boiler) | 36 | 24 | 12 |
| Heating surface used, square feet | 18,136 | 18,136 | 18,136 |
| Pressures (average): | | | |
| Main steam, at boilers, pounds gage | 252.8 | 255 | 243 |
| steam chest, pounds gage | 193.7† | 211.7‡ | 138§ |
| Vacuum in condensers, inches of mercury | 27.5 | 28 | 29.3 |
| Air pressure in fireroom, inches of water | 2.9 | 1.21 | None. |
| Revolutions per minute of main turbines (average): | | | |
| Shaft No. 1 | 855.74 | 688.79 | 397.76 |
| No. 2 | 799.88 | 599.66 | 344.25 |
| No. 3 | 866.79 | 629.06 | 404.62 |
| Mean, all shafts | 840.8 | 639.16 | 382.21 |
| Collective S.H.P., all shafts, from curve | 12,600 | 6,650 | 1,335 |

†M.H.P. ‡I.P.C. §H.P.C.

# U. S. S. *JARVIS.*

## CONTRACT TRIAL PERFORMANCE.

By Henderson B. Gregory, Associate.

An Act of Congress approved June 24, 1910, authorized the construction of six torpedo-boat destroyers, Nos. 37 to 42, inclusive, the *Jarvis* being No. 38 of this number.

She was built under contract by the New York Shipbuilding Company, of Camden, N. J., the price being $640,000, and the time of construction twenty-four months from date of contract, December 3, 1910. She is a triple-screw Parsons turbine vessel, designed to develop 12,000 S.H.P. at about 775 revolutions per minute of the main engines, the contract speed being 29.5 knots and the trial displacement about 742 tons.

In all essential particulars the *Jarvis* is identical with the *McCall*, *Burrows* and *Ammen*, bult by the same company, a full description of which was published in the May, 1911, number of the Journal of the American Society of Naval Engineers, and which applies to this vessel also.

### TRIALS.

The following trials were required:

(*a*) A progressive trial over a measured-mile course for standardizing the screws.

(*b*) A full-speed trial of four hours duration in the open sea at the highest speed obtainable. The average speed to be at least 29.5 knots, with the average air pressure in the fire-rooms not exceeding 6½ inches of water and the steam pressure at the M.H.P. turbine not to exceed 240 pounds above the atmosphere.

(*c*) An endurance and fuel-oil and water-consumption trial of four hours duration in the open sea, at as nearly as possible a uniform speed of 25 knots, the average not to fall below that figure.

(*d*) An endurance and fuel-oil and water-consumption trial at as nearly as possible an average speed of 16 knots.

The various trials were conducted on September 17 and 18, 1912. The standardization trial was run on the Delaware Breakwater measured-mile course, and all other trials were

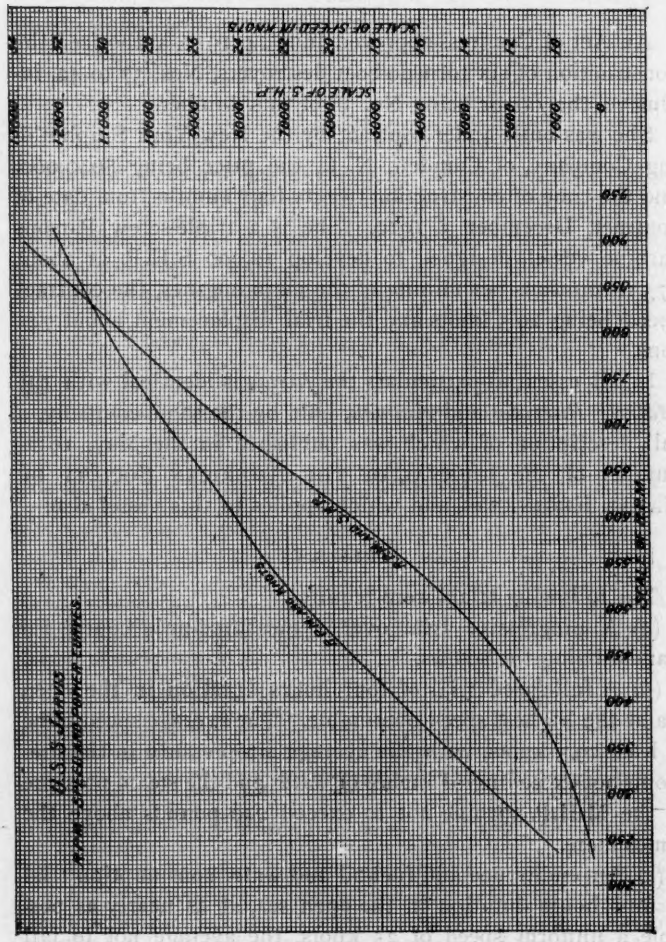



FIG. 1.

## Table I.—Standardization Trial Data, U. S. S. "Jarvis."

Course off Delaware Breakwater. September 17, 1912.

| Number of run..... | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Time on course, min. sec.. | 6 22.8 | 5 18.4 | 5 59.35 | 3 51.25 | 3 38 | 3 51.6 | 3 4.05 | 3 23.25 | 3 3.4 | 2 46.35 | 2 26.4 | 2 49 | 2 13.4 | 2 36.1 | 2 14 | 2 23.7 | 2 7.9 | 2 19.9 | 1 58 | 2 6 | 1 57.8 | 1 59.1 | 1 50.7 | 1 54.9 | 1 51.7 | 1 53.4 |
| Revs. to go 1 knot: | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Shaft No. 1..... | 1,495.2 | 1,279.9 | 1,422.7 | 1,491 | 1,418.6 | 1,517.6 | 1,420.9 | 1,553.9 | 1,404.5 | 1,675.8 | 1,492.5 | 1,706.5 | 1,533.3 | 1,787.9 | 1,540.8 | 1,658 | 1,469.2 | 1,627.7 | 1,563.3 | 1,680.3 | 1,565.9 | 1,731 | 1,649.2 | 1,752.5 | 1,668.4 | 1,747.8 |
| No. 2..... | 1,460 | 1,245.4 | 1,383.4 | 1,268.7 | 1,209.6 | 1,292.5 | 1,196.5 | 1,311.9 | 1,183.8 | 1,429.3 | 1,275.1 | 1,457.5 | 1,302.6 | 1,516.6 | 1,301.6 | 1,601.1 | 1,421.7 | 1,568.7 | 1,469.5 | 1,580.2 | 1,468.9 | 1,610.6 | 1,534.3 | 1,626.6 | 1,549.5 | 1,629.8 |
| No. 3..... | 1,604.2 | 1,354.2 | 1,492.4 | 1,513.7 | 1,446.6 | 1,545 | 1,427.7 | 1,555 6 | 1,407.4 | 1,465 | 1,312.1 | 1,503.1 | 1,330.1 | 1,551.4 | 1,331.7 | 1,634.6 | 1,453.5 | 1,607.1 | 1,573.3 | 1,686.6 | 1,570.1 | 1,764.4 | 1,667.8 | 1,776.1 | 1,687.7 | 1,775.9 |
| Average ..... | 1,519.8 | 1,293.2 | 1,432.9 | 1,424.5 | 1,358.3 | 1,451.7 | 1,348.4 | 1,473 8 | 1,331.9 | 1,523.4 | 1,359.9 | 1,555.7 | 1,389 | 1,618.6 | 1,391.4 | 1,631.2 | 1,448.1 | 1,601.2 | 1,535.4 | 1,649 | 1,535 | 1,702 | 1,617.1 | 1,718.4 | 1,635.2 | 1,717.8 |
| Revs. per minute: | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Shaft No. 1..... | 234.35 | 241.04 | 237.89 | 387.27 | 390.79 | 393.15 | 462.83 | 458.38 | 457.50 | 604 98 | 611.66 | 605.12 | 689.13 | 687.63 | 690.95 | 692.28 | 689.77 | 697.09 | 793.56 | 800.12 | 798.93 | 869.83 | 893.88 | 917.54 | 896.96 | 924.76 |
| No. 2..... | 228.83 | 234 54 | 231.33 | 329.53 | 333.21 | 334.85 | 389.74 | 386.99 | 385.59 | 515.99 | 522.59 | 516.83 | 585.42 | 583.29 | 583.68 | 668.52 | 667.45 | 671.83 | 745.94 | 752.46 | 749.44 | 809.35 | 831.60 | 851.60 | 833.07 | 862.33 |
| No. 3..... | 251.44 | 255.03 | 249.56 | 393.17 | 398.52 | 400.26 | 465.05 | 458.88 | 458.43 | 528.88 | 537.73 | 533.02 | 597.78 | 596.70 | 597.15 | 682.49 | 682.40 | 688.25 | 798.60 | 803.15 | 801.02 | 886.63 | 903.94 | 927.90 | 907.37 | 939.63 |
| Average..... | 238.21 | 243.54 | 239.59 | 369.99 | 374.17 | 376.09 | 439.21 | 434.75 | 433.84 | 549.95 | 577.33 | 551.66 | 624.11 | 622.54 | 623.93 | 681.10 | 697.87 | 685.72 | 779.37 | 785.24 | 783.13 | 855.27 | 876.47 | 899.68 | 879.13 | 908.91 |
| Speed, in knots..... | 9.404 | 11.307 | 10.018 | 15.567 | 16.514 | 15.544 | 19.560 | 17.712 | 19.523 | 21.642 | 24.590 | 21.302 | 26.987 | 23.062 | 26.866 | 25.052 | 28.147 | 25.733 | 30.508 | 28.571 | 30 560 | 30.227 | 32.520 | 31.332 | 32.229 | 31.746 |
| S.H.P., Shaft No. 1.. | 106 | 116 | 120 | 528 | 523 | 536 | 881 | 850 | 860 | 2,162 | 2,232 | 2,193 | 2,818 | 2,863 | 2,894 | 2,594 | 2,500 | 2,560 | 3,188 | 3,233 | 3,150 | 3,900 | 3,920 | 4,091 | 3,956 | 4,169 |
| No. 2.. | 90 | 92 | 91 | 219 | 238 | 239 | 384 | 381 | 380 | 1,093 | 1,119 | 1,082 | 1,687 | 1,667 | 1,711 | 2,716 | 2,712 | 2,788 | 3,490 | 3,540 | 3,507 | 3,927 | 4,055 | 4,363 | 4,165 | 4,375 |
| No. 3.. | 149 | 157 | 147 | 560 | 578 | 590 | 903 | 891 | 378 | 1,118 | 1,149 | 1,165 | 1,674 | 1,678 | 1,672 | 2,548 | 2,581 | 2,621 | 3,590 | 3,572 | 3,542 | 4,139 | 4,486 | 4,149 | 4,257 | 4,501 |
| Total, all shafts..... | 345 | 365 | 358 | 1,307 | 1,339 | 1,365 | 2,168 | 2,122 | 2,118 | 4,373 | 4,500 | 4,440 | 6,179 | 6,208 | 6,277 | 7,858 | 7,793 | 7,969 | 10,268 | 10,345 | 10,199 | 11,966 | 12,461 | 12,603 | 12,478 | 13,045 |
| No. turbines used..... | Five. | | | | | | | | | Four. | | | | | | Three. | | | | | | | | | | |

run in the open sea off the Delaware Breakwater. Good weather prevailed on all the trials.

On the standardization trial twenty-six runs, at various speeds, were made over the measured mile, the data from which were used in plotting the curves shown in Figure 1. The data are given in Table I.

From the official curves the following revolutions per minute of the main engines were found to be necessary to attain the speeds stipulated for the other trials:

| Speed in knots. | R.p.m. |
|---|---|
| 16 | 372.5 |
| 25 | 624.0 |
| 29.5 | 780.0 |

The full-speed and endurance trials were successfully run on the dates noted in Table II. The contract speed was easily exceeded on the full-speed trial, as shown in the table, which also gives the other data obtained on the several trials.

TABLE II.—TRIAL DATA U. S. S. "JARVIS."

| ..... | 4-hour full-power trial. | 4-hour 25-knot trial. | 4-hour 16-knot trial. |
|---|---|---|---|
| Date of trial | 1912. Sept. 18. | 1912. Sept. 18. | 1912. Sept. 17. |
| Speed, in knots | 30.009 | 25.16 | 15.976 |
| Slip of propellers, per cent. of own speed, mean: | | | |
| Starboard propeller | 23.26 | 23.94 | 14.21 |
| Middle propeller | 18.16 | 11.11 | 1.30 |
| Port propeller | 23.32 | 12.85 | 15.04 |
| Draught, mean on trial, feet and inches | 8-$7\frac{5}{16}$ | 8-6 | 8-$6\frac{1}{16}$ |
| Displacement corresponding, tons | 777.3 | 761.2 | 776.4 |
| Number of turbines used | 3 | 4 | 5 |
| boilers used | 4 | 4 | 2 |
| oil burners used (11 per boiler) | 35 to 38 | 21 to 24 | 6 |
| Heating surface used, square feet | 19,200 | 19,200 | 9,600 |
| Pressures (average): | | | |
| Main steam, at boilers, pounds gage | 259 | 264 | 260 |
| steam chest, pounds gage | 189† | 236‡ | 156⁂ |
| Vacuum in condensers, inches of mercury | 27 | 27.8 | 28.25 |
| Air pressure in fireroom, inches of water | 5.7 | 4.4 | 4 |
| Revolutions per minute of main turbines (average): | | | |
| Shaft No. 1 | 819.43 | 693.17 | 390.21 |
| No. 2 | 768.29 | 590.17 | 330.45 |
| No. 3 | 819.98 | 603.41 | 394.03 |
| Mean, all shafts | 802.57 | 628.92 | 371.56 |
| Collective S.H.P., all shafts, from curve | 10,584 | 5,945 | 1,330 |

†M.H.P. ‡I.P.C. ⁂H.P.C.

# PRESENT STATUS OF THE LEBLANC AIR PUMP.

By H. T. Herr, Associate.

The importance of high vacuum appeared with the development of the steam turbine, as has been repeatedly expressed before. Condensation of steam in the absence of air is physically simple, but with the admixture of air, through entrainment in the injection or infiltration, the problem assumes a more difficult aspect. Essentially, then, it must be removed from the steam space in the condenser with the greatest rapidity to obviate the effect its presence exerts upon both the total absolute pressure in the condenser and the rate of heat transference through the tube surfaces (in the case of the surface condenser).

It may be profitable to review the laws which affect condenser operation; mainly, that we keep in mind the reasons for the various designs which have been devised for the maintenance of high vacuum.

First: The total pressure in a confined space is equal and uniform in all parts and directions.

Second: The total pressure of a mixture of two gases, or of a vapor and gas, is equal to the sum of the pressures of the constituents (according to Dalton's Law). In a condenser, for instance, the condition may be expressed as follows:

$$P\,(\text{total}) = P_{steam} + P_{air}.$$

Evidently the greater the accumulation of air and its consequent increasing tension, the higher the total pressure in the condenser and, accordingly, the greater the back pressure on the turbine. Although these are quite obvious facts, they again bring to the fore the necessity of quick removal of the air entering the condenser.

Now, besides the employment of an efficient air pump of ample capacity, cognizance must be taken of the pressure and temperature upon these mixtures where such low tensions prevail. In a paper read before the Institution of Naval Architects, England, Mr. D. B. Morrison presented a special chart which shows these relations in a most instructive way and which has already been reproduced in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. It is recommended for study, and for illustration a few examples may well be presented. The volume of one pound of air with its accompanying water vapor under a pressure of 28.5 inches vacuum will be doubled if the temperature is raised from 70 degrees F. to 82.5 degrees F. If increased to 86 degrees F., the volume to be withdrawn is trebled. Such conditions led to the building of condensers designed for the removal of air at the coldest point of the condenser, and which have become known as counter-current, or contraflo condensers. It is possible only in this way to produce high vacua with the ordinary type of pump which would not be of prohibitive size. From these facts we see that the customary reciprocating air pump, or any other design employing pistons, demands special arrangement of water passes, steam inlet, condensed-water drainage and air offtake.

The Leblanc pump depends upon water as a medium for the ejection of the air. Pumping air by hydraulic process is very old, but all earlier methods applied only the principle of surface friction; that is, a moving jet of water will entrain air by virtue of the existing friction between it and the surrounding air. This principle gives fairly satisfactory results, but the Leblanc pump accomplishes much more than this, it being more efficient, roughly in the ratio of 5 to 1. As will be observed in Fig. 1, a section through the air-pump runner, the water is not projected by the pump as a solid jet, but consecutive sheets, or laminae, are driven across the throat of the pump diffuser, which in themselves constitute small water pistons traveling constantly in a single direction. These pistons are picked up and projected at high

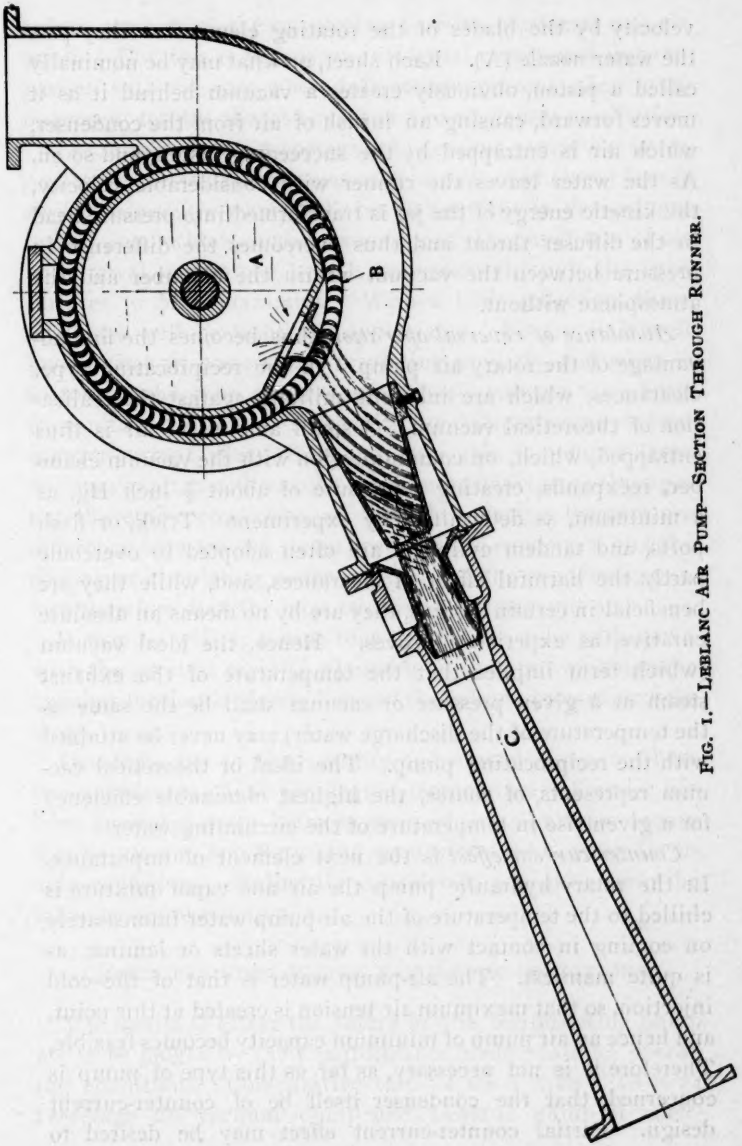



FIG. 1.—LEBLANC AIR PUMP—SECTION THROUGH RUNNER.

velocity by the blades of the rotating element as they pass the water nozzle (A). Each sheet, or what may be nominally called a piston, obviously creates a vacuum behind it as it moves forward, causing an inrush of air from the condenser, which air is entrapped by the succeeding piston, and so on. As the water leaves the runner with considerable velocity, the kinetic energy of the jet is transformed into pressure head in the diffuser throat and thus overcomes the difference in pressure between the vacuum within the chamber and the atmosphere without.

*Avoidance of reversal operation* thus becomes the first advantage of the rotary air pump. In the reciprocating type, clearances, which are inherent, militate against the realization of theoretical vacuum. A small amount of air is thus entrapped, which, on communication with the vacuum chamber, reëxpands, creating a pressure of about $\frac{1}{2}$ inch Hg. as a minimum, as determined by experiment. Trick, or flash ports, and tandem cylinders are often adopted to overcome partly the harmful effect of clearances, and, while they are beneficial in certain designs, they are by no means an absolute curative, as experience proves. Hence, the ideal vacuum (which term implies that the temperature of the exhaust steam at a given pressure or vacuum shall be the same as the temperature of the discharge water) may never be attained with the reciprocating pump. The ideal or theoretical vacuum represents, of course, the highest obtainable efficiency for a given rise in temperature of the circulating water.

*Counter-current effect* is the next element of importance. In the rotary hydraulic pump the air and vapor mixture is chilled to the temperature of the air-pump water immediately on coming in contact with the water sheets or laminæ, as is quite manifest. The air-pump water is that of the cold injection, so that maximum air tension is created at this point, and hence an air pump of minimum capacity becomes feasible. Therefore it is not necessary, as far as this type of pump is concerned, that the condenser itself be of counter-current design. Partial counter-current effect may be desired to

reduce the size of the connection between condenser and air pump. But this is plainly an incidental requirment in comparison with the case where full counter-current effect must be present for the direct benefit of the condenser auxiliaries.

*The third valuable property of the Leblanc system lies in the fact that a more uniform temperature will exist throughout a surface condenser* where full counter-current is derived exteriorly, and which, simultaneously with the rapid rate of air withdrawal, causes the heat transmission through the tube surfaces to be a maximum. With a little reflection, these attributes will be positively appreciated. Firstly, it is well known that air possesses an insulating property, and when it surrounds the tube surfaces it obviously reduces their heat-transmitting capacity. Next, with the pump creating its own counter-current effect, it is not only able to remove large volumes of air, but allows of a maximum temperature throughout the entire condenser, so that the average temperature difference between the steam space of the condenser and the cooling or circulating water is large, and the condensing surface therefore does the most effective work per unit area.

It may be well to notice that the air must be withdrawn with the greatest rapidity where high vacua prevail since the existing tension is always low and the volume proportionately large.

Developments in this country and abroad show that heat transmission per hour per degree temperature difference has been increased from between 300 and 400 B.t.u. per square foot to 500 and 600 B.t.u., and even to 800 B.t.u. under favorable conditions. Ordinarily considered, 500 B.t.u. may be taken as representative.

## CHARACTERISTICS UNDER FAVORABLE CONDITIONS.

The merits of either the rotary or the reciprocating pump are to be judged by their performance under parallel operating conditions. Comparative tests of an $8 \times 18 \times 18 \times 24$ two-stage dry-vacuum pump and Leblanc pump of corre-

sponding capacity have been made, and these show results as exhibited in Fig. 2.

The point to be appreciated from the characteristic curves is that the volumetric capacity of the Leblanc air pump improves as the medium handled becomes more attenuated or

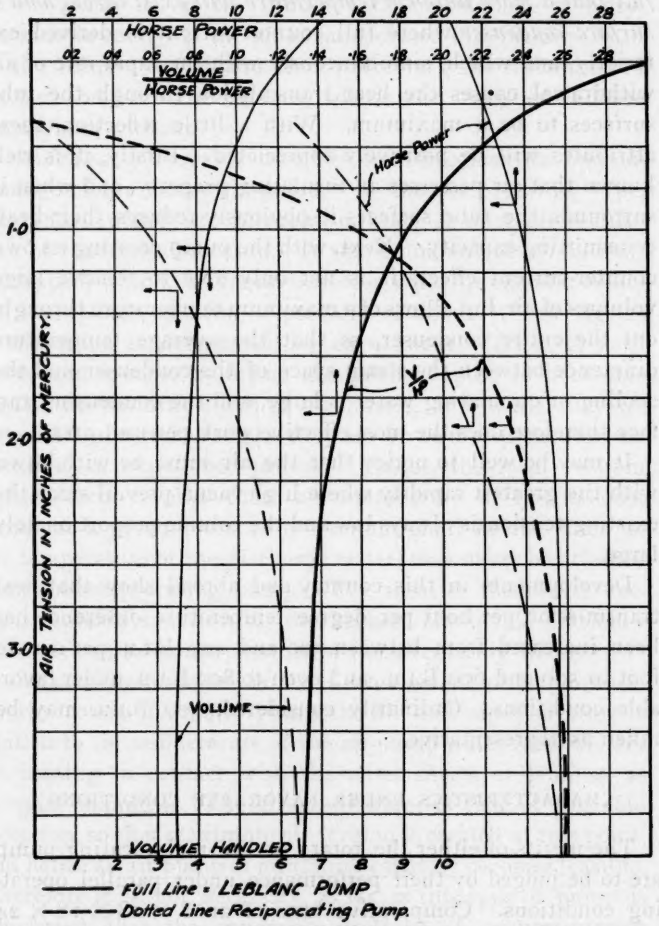



FIG. 2.—COMPARATIVE PERFORMANCES OF LEBLANC AND RECIPROCATING DRY-AIR PUMPS.

rarified, while the reverse is true of the reciprocating type. In the latter there is small possibility of bringing the air tension below ½ inch Hg. absolute. The Leblanc pump, on the contrary, reaches its maximum efficiency below this point, but gradually loses as the pressure rises, its performance remaining virtually constant as the air pressure increases beyond 2 inches Hg. It will be seen that the reciprocating pump has an exactly opposite characteristic. Therefore, where low vacua only are required the Leblanc pump would rarely be selected excepting that it may be desired on account of its mechanical advantages.

In Fig. 2 it will be noted that the volumetric capacity curves coincide at a point corresponding to about 1.5 inches air tension. But at this point the power consumption of the pump is slightly greater than that of the reciprocating pump. However, at approximately 1.15 inches air tension the volumetric displacements per horsepower required by the two types are equal, with a decided advantage in favor of the Leblanc in the high vacuum ranges.

In addition to the above there is another important physical characteristic to be considered, especially in connection with marine installations, viz: the relative weight and floor area of the two types of pump. While reciprocating pumps of the vertical marine type require less weight and floor area than the horizontal construction used in land practice, the Leblanc pump still possesses a distinct advantage in this respect.

It is a simple matter to establish a hot-well pump as a composite part of the air-pump outfit, and in fact the condenser-circulating pump might be just as readily placed on a common shaft, which would effect a consolidated and more compact auxiliary. But this arrangement would obviously lack the flexibility present where separate drive is employed.

## MECHANICAL DESIGN.

It is a self-evident fact that the revolving type of machinery is superior to the reciprocating type, as it obviates all reversal shocks and strains, and, moreover, shows itself to be free from

the complication of parts possessed by the latter and which demand frequent attention to maintain them in good adjustment. The problem of cylinder lubrication is removed from the Leblanc unit. This problem need not be difficult of solution, to be sure, but it may, nevertheless, create trouble where unsuitable vacuum oil is selected and carbonization takes place as a consequence. Turbine drive being usually employed, the exhaust from such an auxiliary is uncontaminated by oily vapors and globules which may prove very objectionable if proper safeguard is not observed both in the initial layout or design and when in service. The Leblanc type is entirely devoid of rubbing surfaces, and should thereby be subject to less wear, its simplicity and accessibility being evident from Fig. 3.

## MARINE APPLICATION.

This type of pump was first developed for land use, and its adoption aboard vessels has necessitated only one important revision. This arose from the fact that it was desirable to circulate sea water through the pump, as the cooler for cooling the fresh water is more efficient and considerably smaller than the air cooler necessary when sea water is not used. It is necessary, for the reasons given heretofore, that the water passing through the Leblanc pump be at the lowest possible temperature, and as the increase in temperature of the water during the cycle is obviously low, relatively little heat is absorbed. However, the continuous passage of water through the pump calls for means of keeping the temperature down to a minimum. For this purpose the air-pump discharge is led back into the base plate or bed plate of the outfit, as shown in Fig. 3, this bed plate being a continuous one from the turbine to the hot-well pump and arranged with tubes for cooling. The rise in temperature of the water going through the air pump is about $\frac{3}{10}$ of one degree F., and as a minimum temperature is essential for the best results, as above noted, the discharge water is cooled down before passing through the air-pump again. The tubes in the bed plate are arranged as

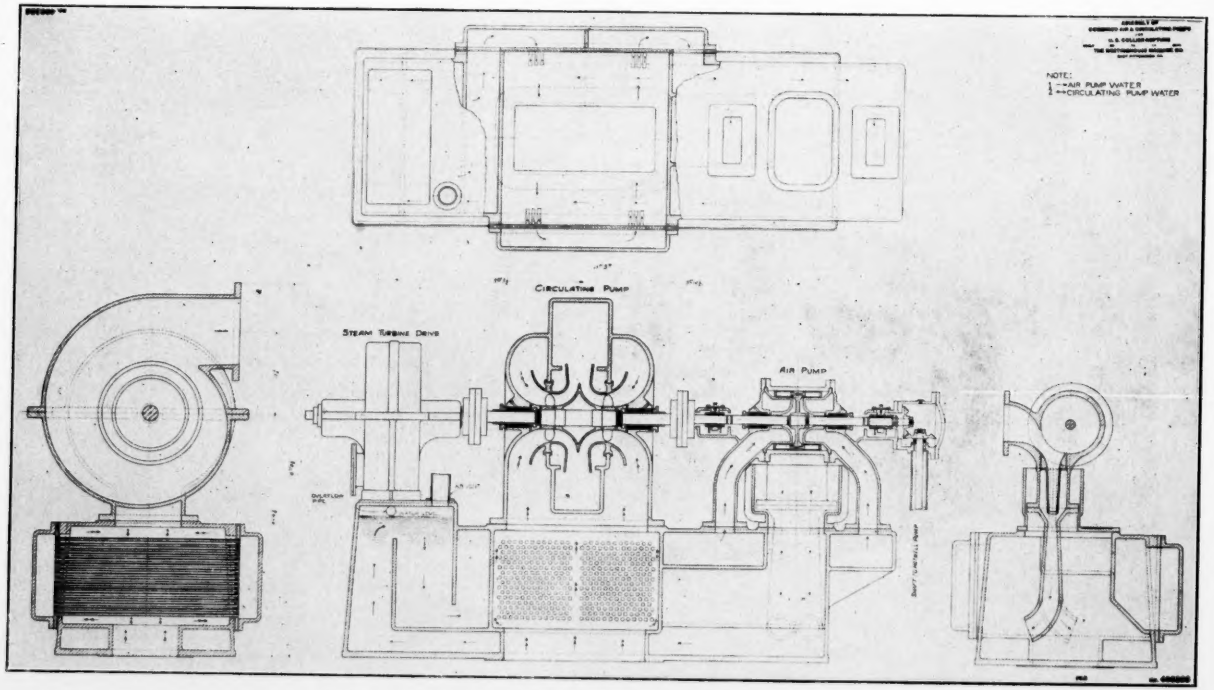



FIG. 3.—IMPROVED MARINE TYPE OF LEBLANC AIR PUMP.

for a surface condenser, all salt water which is used as a cooling medium in the latter being first drawn through the cooler and then pumped through the main condenser. With the large quantity thus passed, maximum cooling of the air-pump water is secured with no sensible increase in the temperature of the water going to the condenser. Obviously no additional power is required for this purpose.

To provide for the escapement of the air entrained in the air-pump discharge one end of the bed plate is arranged with baffles to reduce the velocity of the water, thus allowing ample time for the air to separate out into the space above the overflow pipe and to pass through the air vent shown at the top. As this arrangement only utilizes the bed plate, there is no additional floor area required for its installation, and the weight of the entire unit is increased only a comparatively small amount equal to that of the tubes.

If conditions, that is, space requirements, should demand it the unit is readily adaptable to *vertical-shaft* construction. This would permit the location of the hot-well pump below the turbine cylinder and hence the continuous removal of any condensate collecting at this point. However, the necessity for this provision is quite remote, as the turbine drains are always provided with an ejector, which, in starting, lifts or discharges the water which drains from the turbine by gravity, into the condenser above. The ejector would be omitted in case the turbine drained by gravity into the hot-well pump. When operating, the steam is traveling at sufficient velocity to sweep any moisture collecting, to the condenser. A similar condition obtains not infrequently in stationary work, where the level of the condenser head may be for some reason or other above the turbine exhaust nozzle.

# SIMPLIFIED EQUATIONS FOR THE INDUCTION MOTOR.

By S. M. Robinson, Lieutenant, U. S. Navy, Member.

On account of the adoption of electric propulsion for the collier *Jupiter* it is probable that the service at large will be more or less interested in induction motors. The various methods and diagrams used at present to arrive at the equations of the induction motor seem complicated and difficult to understand, and it is believed that the following solution is much more simple.

The induction motor consists of two main parts, a stator and a rotor. The stator is wound exactly like a polyphase alternator armature, and the rotor may also be wound in this way with its windings short-circuited through slip rings, or it may be simply a drum with parallel copper bars laid in slots on its surface and these short-circuited by a ring of copper on each end joining them all together.

In the stator a rotating field is produced by means of polyphase alternating currents; this field acts on the rotor conductors, generating E.M.F.s and causing currents to flow in them; the reaction of these currents on the rotating field causes the rotor to revolve. The action of the polyphase currents producing a rotating field is shown in Fig. 1. Here we have an iron ring wound with an endless coil, having conductors attached to it at three equidistant points, A, B and C. Suppose these conductors to be supplied with three-phase alternating current; the relation of the three phases at any instant is shown in Fig. 2. Consider the action of the currents at the point o degree on the curves in Fig. 2; here the current entering at A is a maximum, and that leaving at B and C is each equal to one-half A; hence there will be a south

pole at A and two smaller and equal north poles at B and C, which will give a resultant north pole midway between B and C and equal to the south pole at A. Next consider the point 60 degrees on the curves; current is now entering at A and B and leaving at C, and we have a resultant south pole mid-

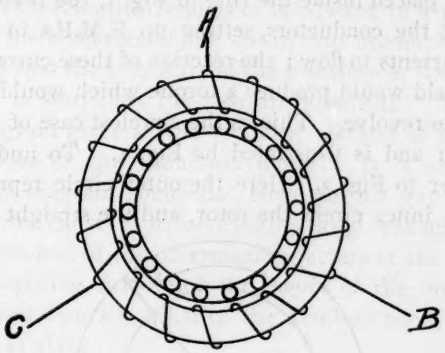



FIG. 1.

way between A and B and an equal north pole at C. At 120 degrees the south pole would be at B and an equal resultant north pole between C and A. At 180 degrees there is a resultant south pole midway between B and C and an equal

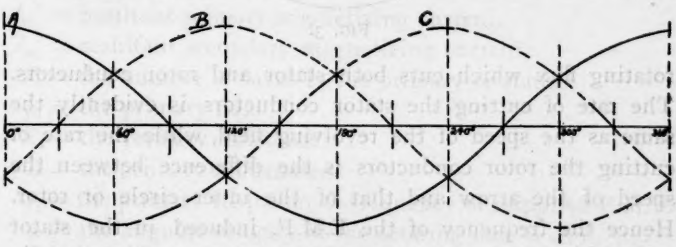



FIG. 2.

north pole at A. Thus we see that the action of the three-phase alternating currents produces a rotating field just the same as if there were real poles on the iron ring and it were revolved mechanically. Also we see that when the south pole has made one-half a revolution the currents have passed

through one-half a period—that is, the south pole goes from A to midway between B and C, while the currents go from o degree to 180 degrees; therefore the angular velocity of the rotating field is the same as that of the currents.

If a cylinder wound with conductors short-circuited at the ends were placed inside the ring in Fig. 1, the revolving field would cut the conductors, setting up E.M.F.s in them and causing currents to flow; the reaction of these currents on the rotating field would produce a torque which would cause the cylinder to revolve. This is the simplest case of the induction motor and is illustrated by Fig. 1. To understand it better refer to Fig. 3. Here the outer circle represents the stator, the inner circle the rotor, and the straight arrow the

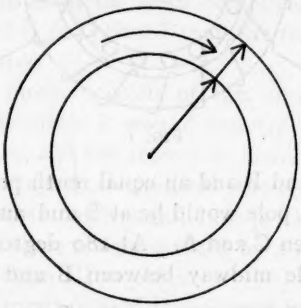


FIG. 3.

rotating flux which cuts both stator and rotor conductors. The rate of cutting the stator conductors is evidently the same as the speed of the revolving field, while the rate of cutting the rotor conductors is the difference between the speed of the arrow and that of the inner circle or rotor. Hence the frequency of the E.M.F. induced in the stator conductors will be the same as that of the supplied E.M.F., while that of the rotor conductors will depend on the speed of the rotor.

The effect of the rotating flux on the stator and rotor of the induction motor is exactly the same as that of the pulsating flux of a transformer on its primary and secondary coils;

in fact, the flux passing through any particular coil does pulsate; the difference between the motor and an ordinary transformer is that, in the latter the frequency of the primary and that of the secondary are the same, while in the former the stator frequency is greater than that of the rotor, but the latter moves mechanically so as to be always in a position to neutralize the reactance of the stator, thus accomplishing the same thing that the secondary of the transformer does. When the motor is stationary the two are identical. The transformer diagram gives us the simplest method of arriving at the equations of the induction motor. It is necessary to modify this diagram somewhat, since the air gap of the induction motor causes it to have such a large leakage flux that this becomes one of the determining factors of the equations. Before proceeding to develop the theory of the motor, using this modified diagram, a list of the symbols used, with their meanings, is given:

$E$ = supplied E.M.F.
$E_1$ = useful primary E.M.F. = $E$ when $R_1$ is zero.
$I$ = primary current.
$I_1$ = primary load current.
$I_m$ = primary magnetizing current.
$I_m'$ = resultant primary magnetizing current.
$I_m''$ = resultant secondary magnetizing current.
$Z_1$ = the number of turns in the primary or stator.
$\varphi_1$ = the primary flux.
$R_1$ = the primary resistance.
$L$ = the primary inductance.
$f_1$ = the primary frequency = frequency of supplied E.M.F.
$\omega$ = the angular velocity of the rotating flux = $2\pi f_1$.
$E_2$ = the useful secondary E.M.F.
$I_2$ = the secondary current.
$R_2$ = the secondary resistance.
$f_2$ = the secondary frequency.
$Z_2$ = the number of turns in the secondary or rotor.
$\varphi_2$ = the secondary flux.
$n$ = the speed of the rotating flux in revolutions per second.

$n_1$ = the speed of the rotor flux in revolutions per second.

$S$ = the slip of the rotor = $\frac{n - n_1}{n}$.

$T$ = the torque of the rotor.

$l$ = the percentage of leakage flux.

$p$ = the number of poles on the stator.

$\theta$ = the angle between $E_1$ and $I$.

$\beta$ = the angle between $E$ and $I$.

$\psi$ = the angle between $I_1$ and $I$.

$\eta$ = the efficiency of the motor.

In Fig. 4 we have, represented by a vector diagram, the conditions in an induction motor when it is treated as a transformer; in this figure the primary resistance, $R_1$, has been

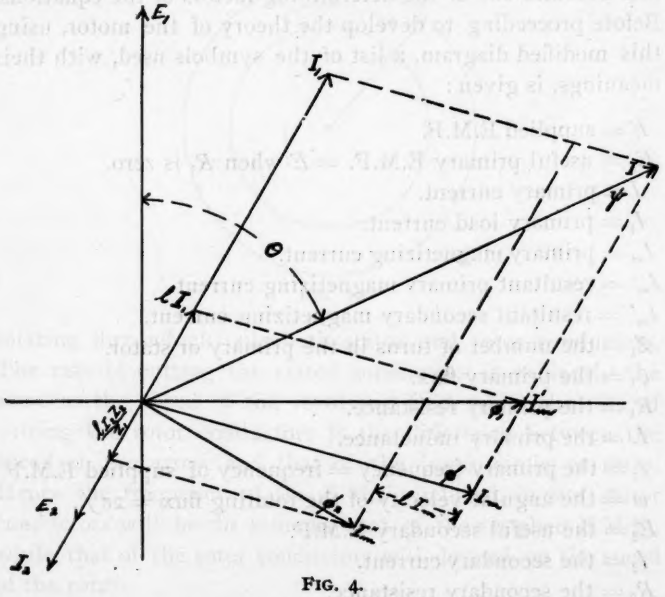



FIG. 4.

neglected, but it will be considered in the next case. As in the transformer, the two load currents, $I_1$ and $I_2$, are exactly opposite to each other in phase; $E_2$ is in phase with $I_2$, since it represents only the useful E.M.F. in the secondary, or that

part which overcomes the resistance of the secondary; the other part would be represented by a vector at right angles to $I_2$, but this part is not considered, as it is not necessary to do so to deduce the equations, as will be shown later. The ampère-turns or fluxes caused by $I_1$ and $I_2$ will not quite balance or neutralize each other, as a certain amount of flux from each coil will pass between the rotor and stator, due to the necessary air gap, and this is called the leakage flux. By reference to Fig. 1 it will be seen that if this flux passes between the rotor and stator it can link with only one of the circuits, and consequently cannot be neutralized by current in the other circuit; hence, it will operate, by its reactance, to cut down the useful E.M.F. in that circuit. Since the flux is proportional to the current the leakage flux will be proportional to $lI_2$ in the secondary. This might also be called the secondary-leakage current. The magnetizing current $I_m$ supplies the flux which induces the E.M.F. to balance $E_1$ in the primary and also to cause $E_2$ in the secondary. It does not supply quite all of this flux, since the leakage flux helps out, as will be seen later. The magnetizing current $I_m$ is itself subject to leakage, and the flux represented by $lI_m$ does not pass through the secondary. In order to get the total resultant flux passing through the secondary it will be necessary to combine the two fluxes represented by $lI_2$ and $(1 - l)I_m$, but in order to reduce the two to the same scale $lI_2$ must be multiplied by the ratio of the number of coils in the secondary to that of the primary, or $\frac{Z_2}{Z_1}$; we can then combine $lI_2 \frac{Z_2}{Z_1}$ with $(1 - l)I_m$ to give the resultant secondary magnetizing current as shown in Fig. 4. In the same way we can combine $lI_1$ and $I_m$ to get the resultant primary magnetizing current. These resultant magnetizing currents must be at right angles to $E_2$ and $E_1$, respectively, since an induced E.M.F. is always 90 degrees in phase behind the current or flux inducing it. This condition makes the trigonometric relations very simple, as most of the triangles are right triangles. The total primary current $I$ is the vector sum of $I_1$ and $I_m$.

Any two E.M.F.s, such as $E_1$ and $E_2$, are proportional to their frequencies, the number of turns in their circuits, and the amount of flux passing through each circuit; hence we have

$$\frac{E_2}{E_1} = \frac{f_2}{f_1} \times \frac{Z_2}{Z_1} \times \frac{\varphi_2}{\varphi_1}.$$

In Fig. 3, we saw that the frequencies were proportional to the relative speeds of cutting conductors in the primary and secondary; that is

$$\frac{f_2}{f_1} = \frac{n - n_1}{n}.$$

The expression $\frac{n - n_1}{n}$ represents the percentage by which the rotor speed differs from synchronous or flux speed, and is called the slip, and denoted by $S$.

Substituting the value of $\frac{f_2}{f_1}$, we have

$$E_2 = E_1 \times \frac{SZ_2}{Z_1} \times \frac{\varphi_2}{\varphi_1}.$$

The two fluxes are proportional to the currents producing them, therefore

$$\frac{\varphi_2}{\varphi_1} = \frac{I_m''}{I_m'}.$$

Since $R_1$ is neglected in this case, the primary reactance $\omega L$, will be the only thing opposing $E_1$, when no load current is flowing in the primary, therefore

$$I_m' = \frac{E_1}{\omega L}. \quad \ldots\ldots\ldots\ldots \quad (1)$$

Since the parts of the primary and secondary load currents which do not produce leakage flux must neutralize each other —that is, have the same number of ampère-turns—we have

$$\frac{I_1(1-l)}{I_2(1-l)} = \frac{Z_2}{Z_1},$$

$$\therefore I_1 = \frac{I_2 Z_2}{Z_1}.$$

From the right triangle we have

$$(I_m'')^2 = (1-l)^2(I_m')^2 - \left[\frac{lI_2Z_2}{Z_1} + l(1-l)I_1\right]^2.$$

Substituting the values of $I_m'$ and $I_1$ already found, we have

$$(I_m'')^2 = \frac{E_1^2(1-l)^2}{\omega^2L^2} - \left[\frac{lI_2Z_2}{Z_1} + \frac{l(1-l)I_2Z_2}{Z_1}\right]^2;$$

$$= \frac{E_1^2(1-l)^2}{\omega^2L^2} - \frac{l^2I_2^2Z_2^2(2-l)^2}{Z_1^2};$$

$$\therefore \quad I_m'' = \sqrt{\frac{E_1^2(1-l)^2}{\omega^2L^2} - \frac{l^2I_2^2Z_2^2(2-l)^2}{Z_1^2}}.$$

Substituting the values of $I_m''$ and $I_m'$ in the flux equation, we have

$$\frac{\varphi_2}{\varphi_1} = \frac{\sqrt{\frac{E_1^2(1-l)^2}{\omega^2L^2} - \frac{l^2I_2^2Z_2^2(2-l)^2}{Z_1^2}}}{\frac{E_1}{\omega L}}.$$

Substituting this expression in the E.M.F. equation, we have

$$E_2 = E_1 \times \frac{SZ_2}{Z_1} \times \frac{\sqrt{\frac{E_1^2(1-l)^2}{\omega^2L^2} - \frac{l^2I_2^2Z_2^2(2-l)^2}{Z_1^2}}}{\frac{E_1}{\omega L}}$$

$$= \frac{SZ_2\omega L\sqrt{\frac{E_1^2(1-l)^2}{\omega^2L^2} - \frac{l^2I_2^2Z_2^2(2-l)^2}{Z_1^2}}}{Z_1}.$$

But $E_2 = R_2I_2$, since the secondary resistance is the only thing $E_2$ has to oppose.

$$\therefore R_2^2I_2^2 = \frac{S^2Z_2^2E_1^2(1-l)^2 - \frac{I_2^2S^2Z_2^4\omega^2L^2l^2(2-l)^2}{Z_1^2}}{Z_1^2};$$

or, $R_2^2 I_2^2 Z_1^4 = S^2 Z_2^2 Z_1^2 E_1^2 (1-l)^2 - I_2^2 S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2$;

or, $R_2^2 I_2^2 Z_1^4 + I_2^2 S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2 = S^2 Z_2^2 Z_1^2 E_1^2 (1-l)^2$;

or, $I_2^2 [R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2] = S^2 Z_2^2 Z_1^2 (1-l)^2 E_1^2$;

$$\therefore I_2 = \frac{S Z_1 Z_2 E_1 (1-l)}{\sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2}} \quad . \quad . \quad . \quad (2)$$

Substituting this value of $I_2$ in the equation for $I_m''$ we have

$$I_m'' = \sqrt{\frac{(1-l)^2 E_1^2}{\omega^2 L^2} - \frac{S^2 Z_2^2 E_1^2 (1-l)^2 (2-l)^2 l^2}{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2}}$$

$$= \sqrt{\frac{E_1^2 R_2^2 Z_1^4 (1-l)^2}{\omega^2 L^2 [R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2]}};$$

$$\text{or } I_m'' = \frac{E_1 R_2 Z_1^2 (1-l)}{\omega L \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2}} \quad . \quad . \quad . \quad (3)$$

Substituting the value of $I_2$ in the equation for $I_1$ we have

$$I_1 = \frac{S Z_2^2 E_1 (1-l)}{\sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2}} \quad . \quad . \quad (4)$$

From the second right triangle we have

$$I^2 (1-l)^2 = (I_m'')^2 + \left[ I_1 (1-l) + \frac{l I_2 Z_2}{Z_1} \right]^2$$

$$= (I_m'')^2 + \left[ \frac{(1-l) Z_2 I_2}{Z_1} + \frac{l I_2 Z_2}{Z_1} \right]^2$$

$$= (I_m'')^2 + \frac{I_2^2 Z_2^2}{Z_1^2}$$

$$= \frac{E_1^2 (1-l)^2 R_2^2 Z_1^4}{\omega^2 L^2 [R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2]} + \frac{S^2 Z_2^4 E_1^2 (1-l)^2}{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2}$$

$$= \frac{E_1^2 (1-l)^2 (R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2)}{\omega^2 L^2 [R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2]}$$

$$\therefore I = \frac{E_1 \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2}}{\omega L \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2-l)^2}} \quad . \quad . \quad (5)$$

The work done is equal to the power supplied less the power lost, which in this case is only the loss in the secondary due to resistance; therefore

$$2\pi n_1 T = E_1 I \cos\theta - R_2 I_2^2.$$

But $$2\pi n_1 = 2\pi n(1 - S)$$

and $$\omega = 2\pi f_1 = 2\pi \times \frac{pn}{2} = \pi p n \text{ or } n = \frac{\omega}{p\pi}$$

$$\therefore 2\pi n_1 = 2\pi(1 - S) \times \frac{\omega}{p\pi} = \frac{2\omega(1 - S)}{p}$$

$$\therefore T\left[\frac{2\omega(1 - S)}{p}\right] = E_1 I \cos\theta - R_2 I_2^2;$$

or $$T = \frac{p}{2\omega(1 - S)}[E_1 I \cos\theta - R_2 I_2^2].$$

But $$\frac{\cos\theta}{\sin\psi} = \frac{I_1(1 - l)}{I_m'} \text{ (from the triangle),}$$

and also $$\sin\psi = \frac{I_m''}{I(1 - l)}$$

$$\therefore \cos\theta = \frac{I_1 I_m''}{I I_m'} = \frac{SZ_2^2 E_1(1 - l)}{\sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}}$$

$$\times \frac{E_1 R_2 Z_1^2 (1 - l)}{\omega L \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}}$$

$$\times \frac{\omega L}{E_1} \times \frac{\omega L \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}}{E_1 \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2}};$$

or

$$\cos\theta = \frac{S^2 Z_1^2 Z_2^2 R_2 \omega L (1 - l)^2}{\sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2} \sqrt{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}} \quad (6)$$

$$\therefore T = \frac{p}{2\omega(1 - S)}\left[\frac{E_1^2 S Z_1^2 Z_2^2 R_2 (1 - l)^2}{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}\right.$$

$$\left. - \frac{E_1^2 R_2 S Z_1^2 Z_2^2 (1 - l)^2}{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}\right]$$

$$= \frac{p}{2\omega(1 - S)}\left[\frac{R_2 S Z_1^2 Z_2^2 (1 - S)(1 - l)^2}{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2}\right];$$

or, $$T = \frac{pE_1^2}{2\omega} \times \frac{R_2 S Z_1^2 Z_2^2 (1 - l)^2}{R_2^2 Z_1^4 + S^2 Z_2^4 \omega^2 L^2 l^2 (2 - l)^2} \quad \ldots \quad (7)$$

To plot curves showing the value of the torque for various speeds it will be necessary to know the value of the leakage factor, the relation of reactance and resistance, and the relation of the number of turns in the primary to the number of turns in the secondary.

Three examples will be chosen to show the effect of varying the rotor resistance.

In all three examples let $Z_1 = Z_2$, and $l = 10$ per cent. $= .1$.

In example (1) let $\omega L = 1{,}000\, R_2$.

In example (2) let $\omega L = 100\, R_2'$.

In example (3) let $\omega L = 10\, R_2''$.

The value of $\omega L$ is supposed to be the same in each, and the rotor resistances $R_2$, $R_2'$ and $R_2''$ to vary.

Substituting these assumed values in the equation for the torque, we have for example (1)

$$(1) \qquad T = \frac{.81\, p E_1^2}{2\, \omega R_2} \times \frac{S}{1 + 36{,}100\, S^2}.$$

To find the maximum value of $T$, differentiate the equation with respect to $T$ and $S$ and equate the first differential to zero, and we have

$$\frac{dT}{dS} = 0 = \frac{1}{1 + 36{,}100\, S^2} - \frac{72{,}200\, S^2}{(1 + 36{,}100\, S^2)^2},$$

or,
$$S = \frac{1}{190} = .0053.$$

Substituting this value of $S$ in the torque equation, we have

$$\text{Maximum } T = \frac{.81\, p E_1^2}{2\, \omega R_2} \times .0026.$$

Proceeding in a similar way for example (2), we have

$$(2) \qquad T = \frac{.81\, p E_1^2}{2\, \omega R_2'} \times \frac{S}{1 + 361\, S^2};$$

$$S \text{ for maximum torque} = \frac{1}{19} = .053;$$

$$\text{Maximum } T = \frac{.81\, p E_1^2}{2\, \omega R_2'} \times .026 = \frac{.81\, p E_1^2}{2\, \omega R_2} \times .0026.$$

For example (3) we have

$$(3) \qquad T = \frac{.81\, pE_1^2}{2\,\omega R_2''} \times \frac{S}{1 + 3.61\, S^2};$$

$$S \text{ for maximum } T = \frac{1}{1.9} = .53;$$

$$\text{Max. } T = \frac{.81\, pE_1^2}{2\,\omega R_2''} \times .26 = \frac{.81\, pE_1^2}{2\,\omega R_2'} \times .026 = \frac{.81\, pE_1^2}{2\,\omega R_2} \times .0026$$

From these results we see that increasing the rotor resistance does not change the value of the maximum torque, but merely alters the speed at which it occurs; consequently, it is possible to arrange the conditions so as to have the maximum torque occur at any desired speed. For example, suppose it is desired to have the maximum torque occur when the motor is at standstill, that is, when $S = 1$.

Substituting this value of $S$ in the torque equation and equating the two members of the denominator, we have

$$.0361\, \omega^2 L^2 = R_2^2,$$

or,

$$\omega L = 5.26\, R_2.$$

The torque curves corresponding to the three examples taken are plotted in Fig. 5. From these it is evident that in motors where the resistance is very low the torque is very small at starting and even less when the motor is running in the opposite direction to the rotating field. To overcome this defect and get a large torque for starting or backing, the rotor conductors are led to slip rings and from there to heavy resistances before they are short circuited, thus giving high initial torque; then, as the motor comes up to speed, the resistance is cut out and the motor runs on a low resistance curve which brings the maximum torque point over near synchronous speed.

This method of varying the rotor resistance can also be used for speed control. Since the speed corresponding to any torque varies with a change of rotor resistance, it is evident that the speed of the rotor can be varied by changing its

resistance; it is not generally used, however, as it is not efficient, the use of the resistance involving large losses.

Since $\omega = 2\pi f$ and $n = \frac{\omega}{p\pi}$, it is evident that a change, either in the frequency of the supplied E.M.F. or in the number of poles on the stator, will change the speed $n$ of the rotating flux and hence of the motor itself; the second method, that of pole-changing, is not generally used for more than two speeds on account of complicating the winding arrange-

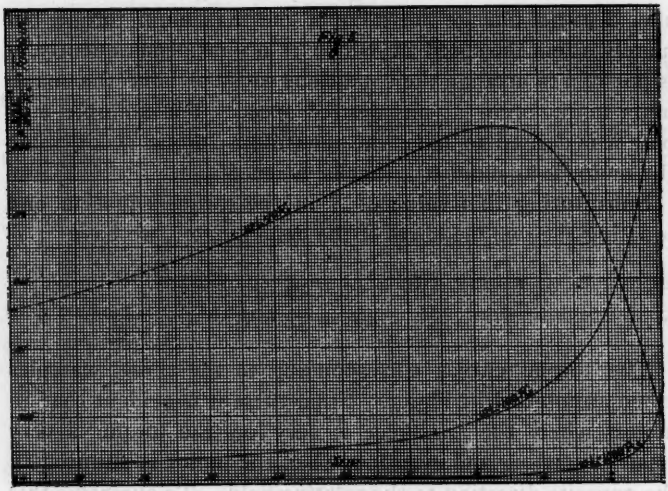


FIG. 5.

ment too much; the first method, however, is the one that is relied upon to get the great range of speed needed for electric ship propulsion.

Since the torque varies as the square of the voltage, it is evident that any change in the supplied E.M.F. would result in a change of speed for any given torque; this would also enter into speed changes for ship propulsion to a slight degree, since the voltage will be varied as well as the frequency by changing the speed of the generator.

In the equations already deduced, the primary or stator

resistance, $R_1$, has been neglected, but it must be taken into account to get the final form of the equations. Fig. 6 shows the change this element makes in the vector diagram. The E.M.F. component necessary to account for the $RI$ drop of

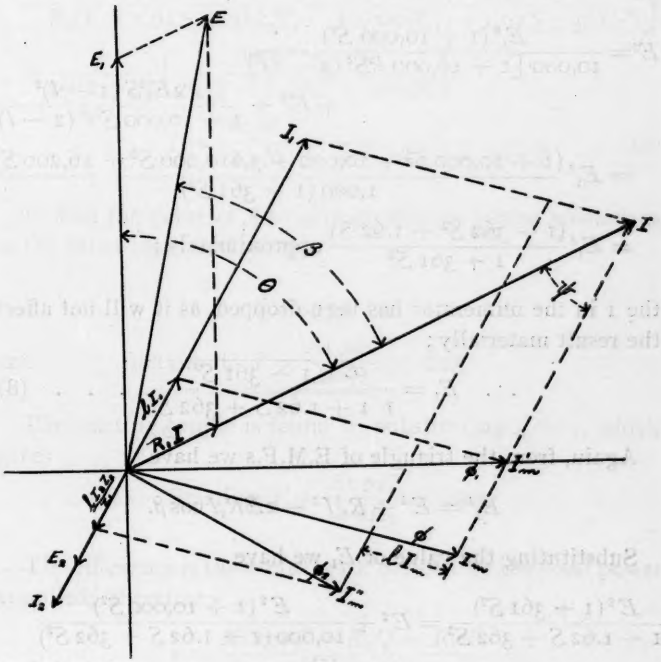



FIG. 6.

potential is compounded vectorially with $E_1$ to give $E$, which is the E.M.F. that must be supplied. To simplify the equations, they will be deduced for a motor with all conditions given. The conditions assumed will be

$$Z_1 = Z_2;$$
$$R_1 = R_2;$$
$$\omega L = 100\, R_2.$$

From the triangle of E.M.F.s we have

$$E^2 = E_1^2 + R_1^2 I^2 + 2E_1 R_1 I \cos\theta.$$

Substituting the values of $E_1$, $I_2$, $I$ and $\cos\theta$ already found, and also substituting the assumed values of $Z_1$, $Z_2$, $R_1$ and $R_2$, we have

$$E^2 = \frac{E_1^2(1 + 10{,}000\,S^2)}{10{,}000\,[1 + 10{,}000\,l^2S^2(2 - l)^2]} + E_1^2 + \frac{2E_1^2S(1 - l)^2}{1 + 10{,}000\,S^2l^2(2 - l)^2}$$

$$= E_1^2\frac{(1 + 10{,}000\,S^2 + 10{,}000 + 3{,}610{,}000\,S^2 + 16{,}200\,S)}{1{,}000\,(1 + 361\,S^2)}$$

$$= E_1^2\frac{(1 + 362\,S^2 + 1.62\,S)}{1 + 361\,S^2} \text{ approximately;}$$

the 1 in the numerator has been dropped, as it will not affect the result materially;

$$\therefore \quad E_1 = \frac{E\sqrt{1 + 361\,S^2}}{\sqrt{1 + 1.62\,S + 362\,S^2}}. \quad . \quad . \quad . \quad (8)$$

Again, from the triangle of E.M.F.s we have

$$E_1^2 = E^2 + R_1^2I^2 - 2ER_1I\cos\beta.$$

Substituting the value of $E_1$ we have

$$\frac{E^2(1 + 361\,S^2)}{1 + 1.62\,S + 362\,S^2} = E^2 + \frac{E^2(1 + 10{,}000\,S^2)}{10{,}000\,(1 + 1.62\,S + 362\,S^2)} - \frac{2\,E^2\cos\beta\,\sqrt{1 + 10{,}000\,S^2}}{100\,\sqrt{1 + 1.62\,S + 362\,S^2}};$$

$$\therefore \cos\beta\,(\text{power factor}) = \frac{.01 + 81\,S + 100\,S^2}{\sqrt{1 + 10{,}000\,S^2}\,\sqrt{1 + 1.62\,S + 362\,S^2}}. \quad . \quad (9)$$

Since the work done is equal to the power supplied less the losses, which include both stator and rotor resistance losses in this case, we have

$$2\pi n_1 T = EI\cos\beta - R_2I_2^2 - R_1I^2.$$

Substituting the values already found, we have

$$T = \frac{p}{2\omega(1-S)}\left[\frac{E^2(.01+81S+100S^2)}{100R_2(1+1.62S+362S^2)} - \frac{.81S^2E^2}{R_2(1+1.62S+362S^2)} - \frac{E^2(1+10{,}000S^2)}{10{,}000R_2(1+1.62S+362S^2)}\right]$$

$$= \frac{p}{2\omega(1-S)}\left[\frac{.81E^2(1-S)}{R_2(1+1.62S+362S^2}\right],$$

$$\therefore\quad T = \frac{.81pE^2}{2\omega R_2} \times \frac{S}{1+1.62S+362S^2}. \quad . \quad (10)$$

To find the value of $S$ to give maximum torque proceed as in the other cases, and we get

$$S = .053,$$

and $$\text{maximum } T = \frac{.81pE^2}{2\omega R_2} \times .025.$$

The starting torque is found by substituting $S = 1$, which gives

$$\text{Starting } T = \frac{.81pE^2}{2\omega R_2} \times .003.$$

The efficiency is the useful work divided by the total power supplied; therefore

$$\eta = \frac{EI\cos\beta - R_2I_2^2 - R_1I^2}{EI\cos\beta}$$

$$= \frac{\dfrac{.81SE^2(1-S)}{R_2(1+1.62S+362S^2)}}{\dfrac{E^2(.01+81S+100S^2)}{100R_2(1+1.62S+362S^2)}}$$

$$\therefore\quad \eta = \frac{81S(1-S)}{.01+81S+100S^2}. \quad . \quad . \quad . \quad . \quad . \quad (11)$$

To find $S$ for maximum efficiency proceed as in the torque equation, and we get

$$S = .0074$$

and maximum $\eta = 96.7$ per cent.

The curves of torque, power factor and efficiency corresponding to equations 9, 10 and 11 are plotted in Fig. 7.

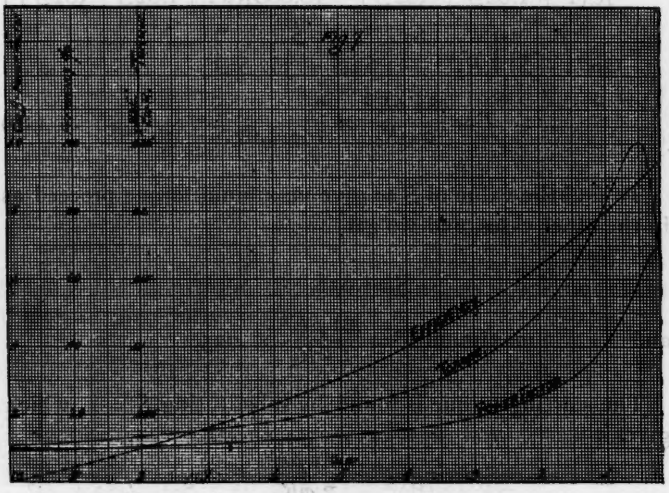


FIG. 7.

In arriving at these equations, the hysteresis and eddy current losses have been neglected, but these would not modify the form of the equations.

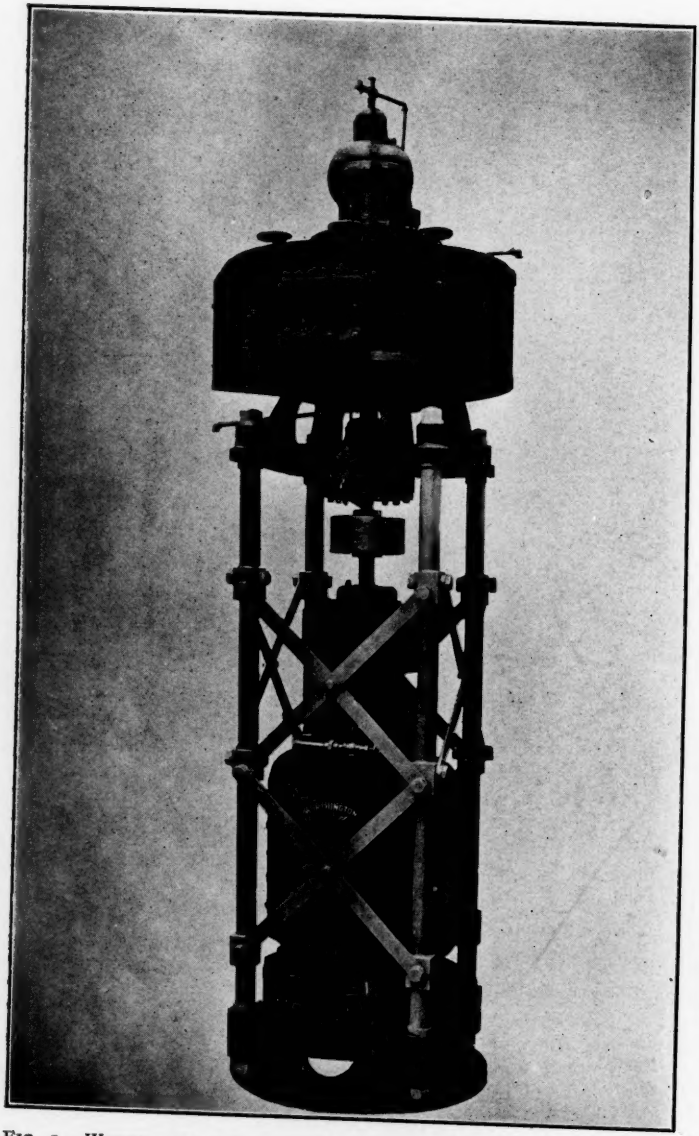


FIG. 1.—WORTHINGTON CENTRIFUGAL BOILER-FEED PUMP WITH TERRY TURBINE DRIVE.

# TEST OF A WORTHINGTON CENTRIFUGAL TURBINE-DRIVEN FEED PUMP AT THE NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

## OBJECT OF TEST.

The object of the test was to determine the general performance and economy of the machine with regard to its use as a boiler-feed pump. This included: The relative merits of governing with constant speed and with constant water-discharge pressure; and the effects, with different rates of pumping water, of varying the discharge-water pressure, the temperature of the water pumped, the number of steam nozzles open, and the exhaust steam pressure.

## DESCRIPTION OF MACHINE TESTED.

The machine tested comprises a Worthington vertical four-stage centrifugal pump driven by a Terry steam turbine. The turbine is supported above the pump on standards and is connected to the pump by a flexible coupling. The machine is shown in figure 1.

The pump consists of a casing containing various water passages and in which revolve four impellers. Both casing and impellers are of composition, and the steel shaft is incased in a composition sleeve when in contact with water. The shaft passes through a stuffing box at each end of the casing. The upper box at the high-pressure discharge end is provided with a drain connection to the suction pipe, the valve in which may be opened when necessary to relieve the water pressure on the packing. The lower box at the low-pressure suction end is provided with a drain.

The shape of the water passages and of the impellers may be seen by referring to figure 2. The cover forming one-half

the casing is shown at the right. The shaft with impellers occupies, of course, a vertical position in the casing, the left end being uppermost. It will be noted that each impeller has a double suction. The water from the suction pipe enters the lowest impeller at both sides and around the shaft. Within the impeller it is thrown radially outward by centrifugal force and its pressure is increased by the revolving vanes. From the outer edge of the impeller the water enters the diffusion vanes which guide the water to the passages leading to the two suction openings of the second impeller. The water passes in a similar manner through the other impellers and thence to the discharge pipe.

The difference in pressure produces water leakage from outer discharge edge to inner suction edge of each impeller and from the suction of one impeller to the suction of the adjacent impeller. This leakage is reduced by labyrinth packing rings. The packing ring between adjacent impellers is rigidly fixed to the shaft and revolves in the corresponding groove to be seen in the casing, figure 2. The packing ring on the suction edge of each impeller has a neat running fit on the impeller and is held stationary in the corresponding groove in the casing. One of these packing rings has been removed from the right-hand impeller in figure 2 and hung between it and the adjacent impeller.

The shaft is held in place by two bearings, one above and the other below the casing. Between the lower stuffing box and bearing a collar is provided to throw off by centrifugal force any water which may leak through the stuffing box. Each bearing contains two half-boxes lined with babbitt.

The weight of the rotating parts and any unbalanced water pressure on the impellers is borne by a thrust bearing just above the upper shaft bearing. The symmetrical construction of the impellers, however, reduces the unbalanced water pressure to a minimum. The construction of this thrust bearing may be seen in figure 2. It consists essentially of two ball-bearings, each bearing consisting of a series of balls confined between two races. The lower ball-bearing cares for the down-

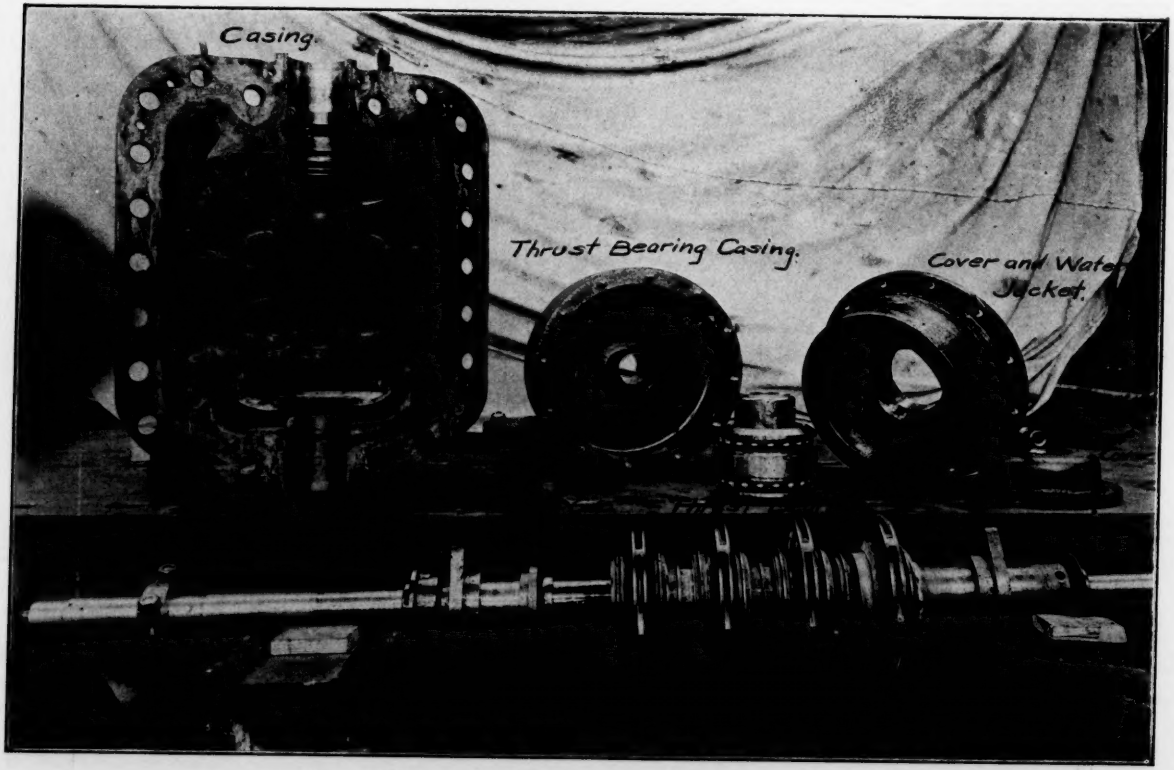



FIG. 2.—WORTHINGTON CENTRIFUGAL BOILER-FEED PUMP.

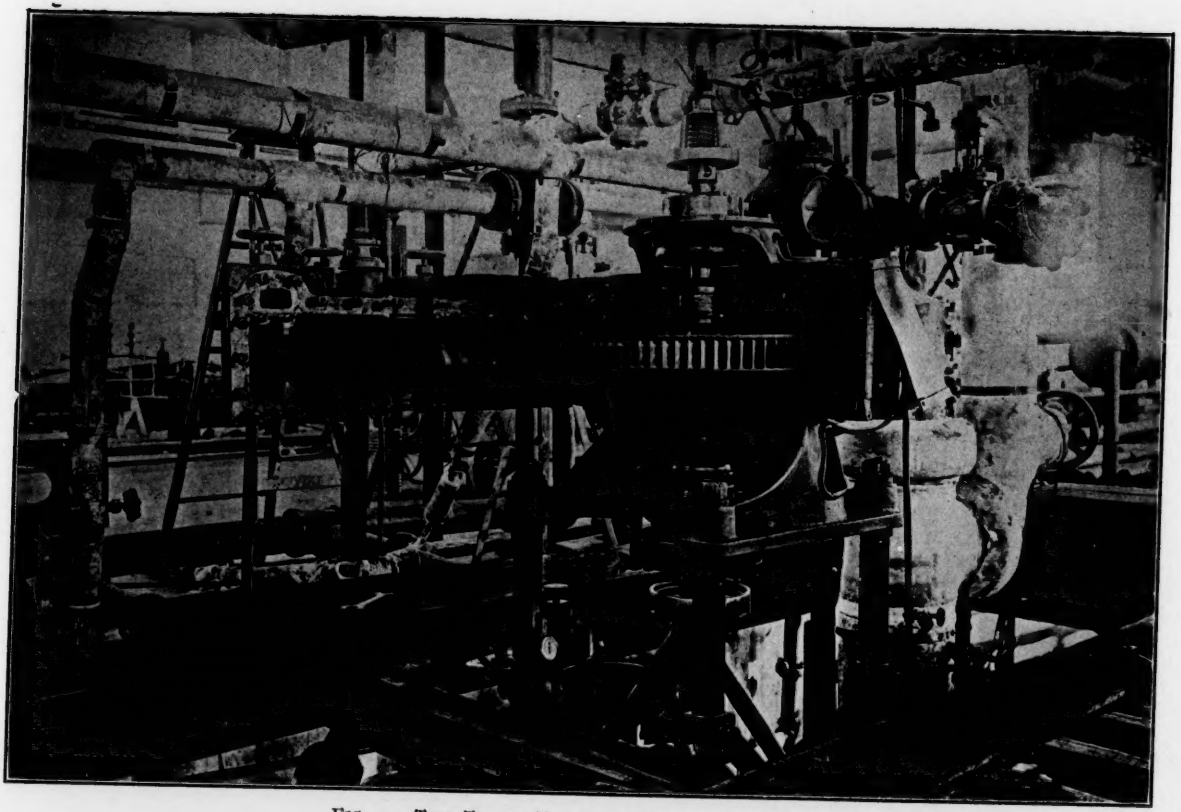


FIG. 3.—THE TERRY TURBINE USED IN THE TESTS.

ward thrust and the upper ball-bearing for the upward thrust, the intermediate cylindrical part being held on the shaft between the shoulder and collar seen at the upper end. The lower ball-bearing is supported on its spherical seat in the casing, and the upper-ball bearing is held down by a spherical seat on the cover. Suspended from the cover is the hollow annular box through which cooling water is circulated. The split collar is not bolted to the shaft until the whole thrust bearing is assembled with main cover: then this collar is protected by a second cover bolted to the top of the main cover.

The ball-thrust-bearing is lubricated by running in a bath of oil. The other bearings must be supplied with sight-feed oil cups.

The interior of the steam turbine is revealed in figure 3. One-half of the casing has been swung open upon a hinge at the left and the covering of the governor has been removed. The two halves are held together by a number of through bolts. Steam is conducted from the steam connection on one half-casing to the nozzles in the other half-casing through the small rectangular openings seen in the extreme upper corners.

The rotor consists of a single wheel 24 inches in diameter, made up of two steel discs bolted together over a steel hub. The buckets are made of steel punchings fitted into grooves cut in the discs. The steam is directed onto the wheel by eight nozzles, four of which are provided with valves. Each nozzle is fitted with four buckets on the side of the nozzle in the direction of rotation. The steam is fully expanded in the nozzle, leaving it at a high velocity. The shape and arrangement of the wheel and nozzle buckets cause the steam to take a helical path, its velocity continually decreasing until it is discharged into the space within the casing where the pressure is determined by the exhaust connections. The casing is provided with a relief valve to be set at some predetermined pressure; in this case about 12 pounds per square inch gage.

Steam-sealing glands are placed on the shaft where it passes through the casing, and consist of solid rings with water-

packing grooves. The glands are of composition and pass through solid cast-iron rings fitting tightly in the casing. Each gland is free to turn within the cast-iron ring and around the shaft, except for the drain pipe screwed into the outside flange and held by an iron strap to one of the ribs on the casing. Composition collars are attached to the shaft just beyond the sealing glands to prevent water entering the bearings or oil entering the turbine.

At the lower end of the shaft is located a shaft bearing, a ball-thrust-bearing and an oil pump. The oil pump consists of a single-threaded worm with semi-circular grooves revolving in a closely fitting cylinder. The pitch of the thread is in such direction as to raise the oil from the bottom to the top of the cylinder where two openings are placed—one leading to the oil-supply pipe to the bearings and governor, and the other leading to a relief valve to regulate the oil-discharge pressure. The oil is returned from the bearings to the pump. The bottom of the pump casing is cored for cooling-water circulation.

At the upper end of the shaft is the second shaft bearing and the governor mechanism. The latter consists essentially of two cylindrical bars pivoted at their lower ends and held with their upper ends pressed inward towards the shaft by a helical spring wound around the shaft. Centrifugal force causes these bars to fly outward against the spring pressure and in so doing raise a spindle passing through the hollow shaft. At the top end of this spindle is a ball on which rests the cap carrying one end of the governor lever. By this mechanism an increase of speed lowers the governor-valve spindle.

The governor valve is of the double-poppet type, the lower disc of which is cylindrical and has a conical seat while the upper disc is conical and has a seat on the edge of a cylindrical hole. As the steam enters between the two discs its pressure acts beneath the upper disc to open it and above the lower disc to close it. The object is to provide a balanced valve which may be tightly closed.

The flexible coupling between the turbine and the pump

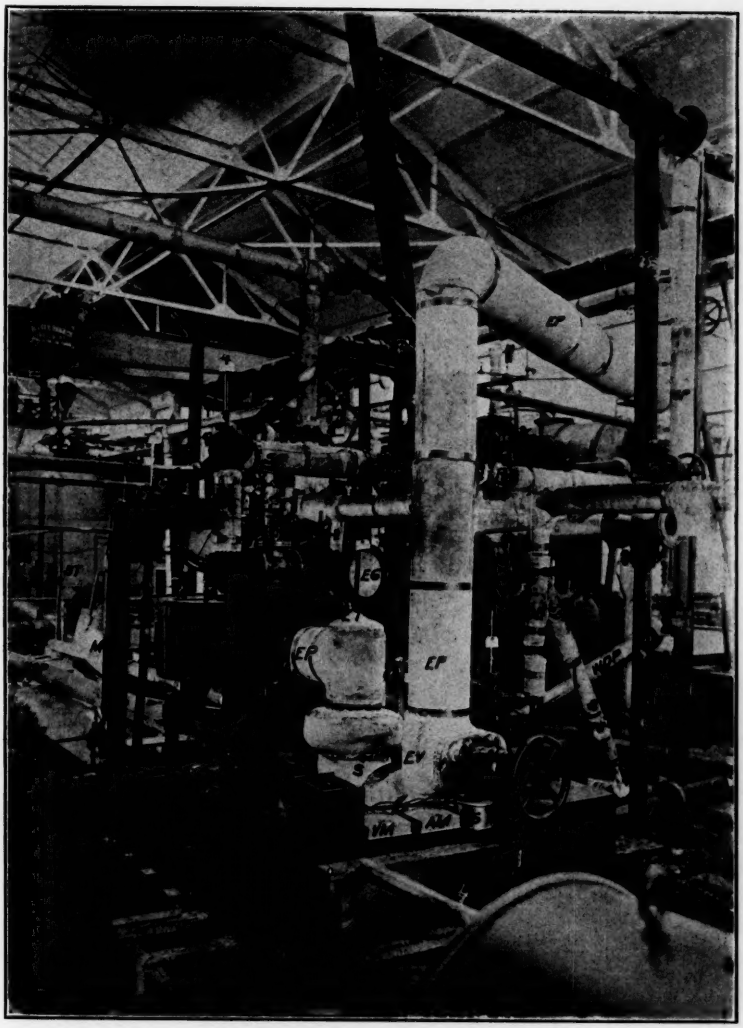



FIG. 4.—WORTHINGTON CENTRIFUGAL BOILER-FEED PUMP. GENERAL ARRANGEMENTS FOR TEST.

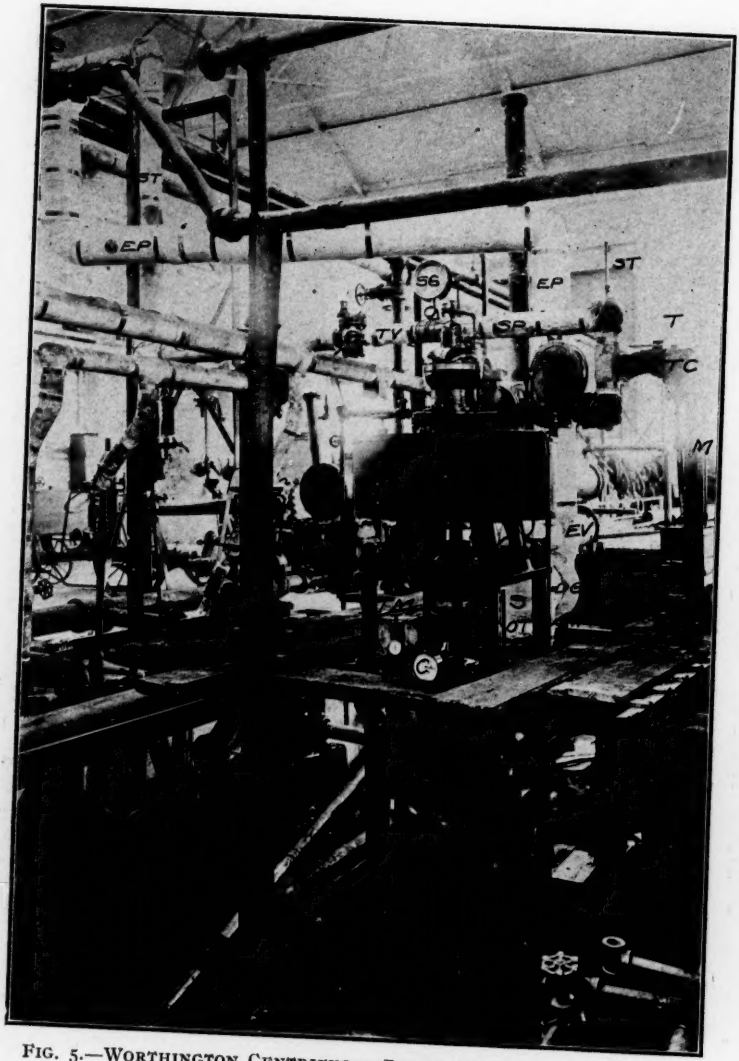



FIG. 5.—WORTHINGTON CENTRIFUGAL BOILER-FEED PUMP. GENERAL ARRANGEMENTS FOR TEST.

consists of two flanges with pins projecting from one flange into rubber-bushed holes in the other flange. Each flange is fastened by key and set screw to its shaft.

## GENERAL ARRANGEMENT FOR TEST.

The general arrangement of the apparatus for the test is shown in figures 4, 5 and 6.

Steam was supplied the turbine through the steam pipe SP, its pressure being reduced and regulated to the desired value by the regulator R when the throttle valve TV was opened wide. During the first six runs the steam passed directly to the governor valve on the turbine; but during the remaining runs the steam passed through the Foster pump governor F, which still further reduced the steam pressure according to the load. The discharge pressure of the centrifugal pump acted upon the Foster pump governor through the $\frac{1}{4}$-inch vertical pipe leading from the top of the governor to the discharge pipe.

The temperature and pressure of the steam supplied were measured by means of the steam thermometer ST and steam gage SG, respectively. The quality was determined by the throttling calorimeter TC supplied with thermometer T and mercury gage M.

The exhaust steam passed through the exhaust pipe EP, separator S, and exhaust valve EV to a condenser, not shown Two scales and tanks were provided for weighing the condensed steam. A drain from the bottom of the separator was carried to the bottom of a tank on scales TS seen in the pit in figure 5. The hot water separated from the exhaust steam was cooled by mixing with the water in the tank, thus preventing evaporation. The drains DG from the steam glands of the turbine also lead to this tank.

The temperature and pressure of the exhaust steam were measured by means of the exhaust thermometer ET and the exhaust-pressure gage EG. The quality was determined by means of the separator S and the electric calorimeter EC. The watts supplied this electric calorimeter were regulated by

the water rheostat WR, and indicated on the voltmeter VM, with multiplier M, and the ammeter AM, with shunt S. The sampling pipe for the electric calorimeter consisted of a ½-inch pipe extending across the exhaust pipe below the exhaust valve EV. It was nearly closed at the far end and had a number of ⅛-inch holes drilled throughout its length. The outlet OE from the electric calorimeter EC entered the exhaust pipe above the exhaust valve EV, the drop in pressure through the valve causing the steam flow through the calorimeter. The temperature of the steam leaving the electric calorimeter was shown on thermometer T. A mercury gage was also connected to the outlet pipe of the calorimeter.

The water pumped by the centrifugal pump flowed from the suction tank ST (see figure 4) through the water suction pipe WSP and the suction valve SV (see figure 6). The static pressure of the water between the suction valve SV and the pump was measured by the mercury gage MG. The temperature of the suction was measured by a thermometer in the supply pipe.

The water was discharged through the discharge valve VD and the water discharge pipe WDP to a pit not shown, or to the suction tank again. On the way, the water flowed through a venturi meter and a feed-water heater FH. The static pressure of the water discharged from the pump was indicated on the discharge gage DG.

Cooling water was supplied the pump thrust bearing through a ½-inch supply pipe CP containing a water meter. The outlet temperature was measured by thermometer OT.

The revolutions of the machine were measured by the counter C, to operate which a worm wheel was designed and fitted to indicate one revolution on the counter for every ten revolutions of the shaft. The tachometer TM was also applied to show the speed at any instant.

## CONDUCT OF TEST.

The first six runs were made to determine the operation of the machine when controlled by the governor valve of the turbine. As the turbine was marked to be supplied with steam

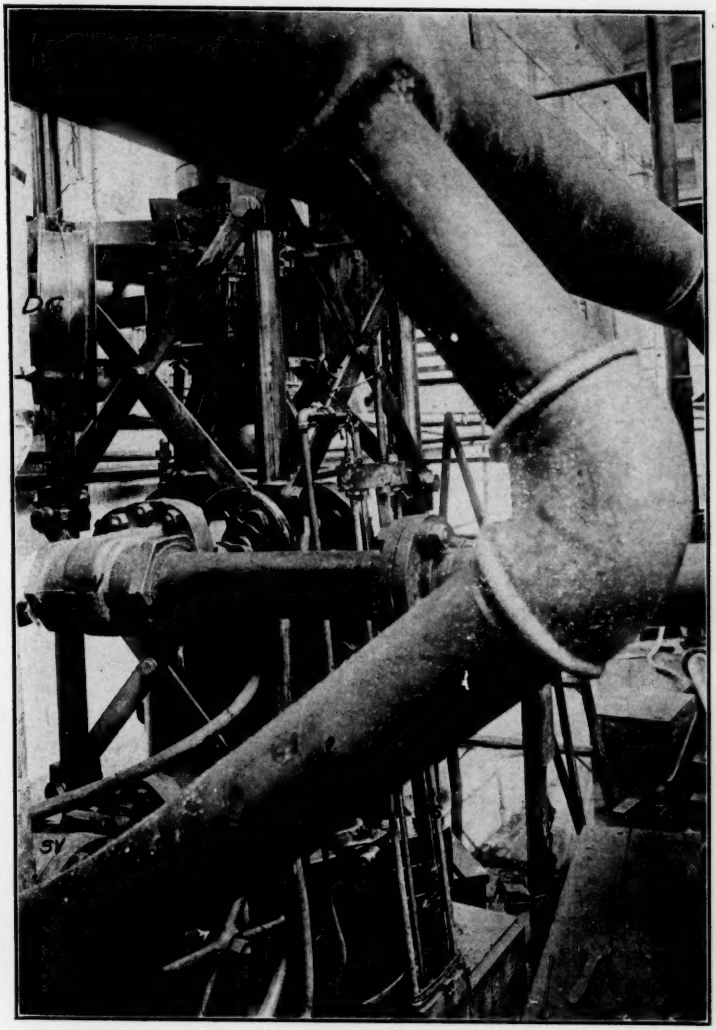


FIG. 6.—WORTHINGTON CENTRIFUGAL BOILER-FEED PUMP. GENERAL ARRANGEMENTS FOR TEST.

at 210 pounds per square inch gage pressure, this value was used during these runs. The exhaust pressure was maintained at about 5 pounds per square inch gage. The governor was set for a speed of about 2,800 r.p.m. The rate of discharge of the pump was regulated by discharge valve DV and other valves in the discharge line.

Runs 7 to 30, inclusive, were made with the machine under the control of the Foster pump governor. Runs 7 to 11, inclusive, were made with a discharge pressure of about 274 pounds per square inch gage; runs 12 to 29, inclusive, with a discharge pressure of about 316 pounds per square inch gage; and run 30 with a discharge pressure of 340 pounds per square inch gage, to show the effect of the variation of the discharge pressure upon the economy. The steam pressure was maintained at about 250 pounds per square inch gage, since the first series of runs had shown the pump to be suitable for this boiler pressure. Run 30, however, was made with a steam pressure of 200 pounds per square inch gage and less to find the effect of a drop in steam pressure. The exhaust pressure was maintained at about 5 pounds per square inch gage except during runs 23 and 24, when it was changed to 1 and to 9 pounds per square inch gage, respectively. Runs 25 to 29 were made with water at about 170 degrees F., and runs 23, 24 and 30 were made while the water was being warmed, to save time. Runs 12 to 22 were made with 4, 5 or 8 nozzles open to determine the effect of so operating the turbine.

The amount of water pumped was determined by noting the fall of the water level in the suction tank ST and by reading the venturi meter. During runs 1, 2, 7, 8, 12, 13, 14, 15, 25 and 26 the level of the water in the suction tank was allowed to fall throughout the run. During runs 23, 24, 27, 28, 29 and 30 the water was simply circulated through the suction tank, the centrifugal pump, the feed-water heater and the venturi meter to avoid loss of warm water, and no fall in the water level in the suction tank occurred. During the remaining runs the water was not circulated; its level in the suction

tank was allowed to fall for various intervals during the run and the drop in level noted. A duplex fire pump served to fill up the tank between the intervals of drop in water level.

All runs were of 60-minute duration. The indication of the speed counter, the level of the water in the suction tank, the weight of the condensed exhaust steam, the weight of water separated therefrom, and the indication of the cooling-water meter were read exactly at the end of each 5-minute interval. The venturi meter and the various temperatures and pressures were read immediately thereafter, with the exception of the barometer, which was read at the beginning and end of each run.

A sample of the water pumped was secured during each run and was later tested for density.

The electric calorimeter on the exhaust-steam pipe was operated as follows: Immediately after taking a reading the water rheostat WR was changed to give a new rate of supplying energy to the steam. Then, at the time of the next reading, the calorimeter had had 5 minutes to reach a state of thermal equilibrium.

The condenser was found to be without leak by the silver nitrate test. The scales were calibrated and found accurate. The thermometer and gages were calibrated before and after the test.

## OBSERVATIONS AND DEDUCED RESULTS.

The main observations and results deduced therefrom are given in Table I. Runs 1 to 6, inclusive, were made under the control of the turbine governor; runs 7 to 30, inclusive, under the control of the Foster pump governor. Runs 7 to 11, inclusive, were made with about 274 pounds per square inch gage water-discharge pressure; runs 12 to 29, inclusive, with about 316 pounds per square inch gage water-discharge pressure; and run 30 with 340 pounds per square inch water-discharge pressure. Runs 12 to 22, inclusive, were made with various numbers of nozzles in use, runs 23 and 24 with different exhaust pressures than the other runs, and runs 25 to 29,

TABLE I.—GENERA

| Run Number. | Date. August, 1912. | Duration of Run, Minutes. | Barometer. Corrected, In. of Mercury. | Barometer, Lb. per Sq. In. | Room Temperature, Degrees Fahr. | Total Number of Revolutions. | Average Speed, Revolutions per Minute. | Steam Used, Lb. per Hour. | Number of Nozzles Open. | Steam Used per Nozzle Lb. per Hour. | Steam Pressure at Turbine, Lb. per Sq. In. Gage. | Steam Pressure at Turbine Lb. per Sq. In. Absolute. | Steam Temperature at Turbine, Degrees Fahr. | Calorimeter Temperature, Degrees Fahr. | Calorimeter Pressure, Inches of Mercury. | Moisture, Per Cent. | Heat in Steam above 32° F. B.T.U. per Lb. | Steam Pressure at Nozzles, Lb. per Sq. In. Gage. | Steam Pressure at Nozzles Lb. per Sq. In. Absolute. | Superheat at Nozzle, Degrees Fahr. | Moisture at Nozzle, Per Cent. | Exhaust Pressure at Turbine, Lb. per Sq. In. Gage. | Exhaust Pressure at Turbine, Lb. per Sq. In. Absolute. | Exhaust Temperature at Turbine, Degrees Fahr. | Heat in Water at Exhaust Temperature above 32°, B.T.U. | Heat in Live Steam above Exhaust Temperature, B.T.U. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 |
| 1 | 12 | 60 | 29.94 | 14.70 | 89.5 | 156750 | 2613 | 2596 | 5 | 519 | 209.3 | 224.0 | 391.2 | 293.8 | 10.07 | 1.49 | 1187.4 | 98.6 | 113.3 | — | 0.12 | 4.4 | 19.1 | 224.9 | 193.0 | 994 |
| 2 | 12 | 60 | 29.93 | 14.70 | 85.4 | 156500 | 2608 | 2971 | 5 | 594 | 209.1 | 223.8 | 391.5 | 293.5 | 10.30 | 1.39 | 1188.2 | 113.0 | 127.7 | — | 0.27 | 4.4 | 19.1 | 224.5 | 192.6 | 995 |
| 3 | 13 | 60 | 30.00 | 14.73 | 88.1 | 156000 | 2600 | 3321 | 5 | 664 | 205.9 | 220.6 | 390.2 | 298.0 | 9.90 | 1.22 | 1189.4 | 131.6 | 146.3 | — | 0.42 | 4.4 | 19.1 | 225.2 | 193.3 | 996 |
| 4 | 13 | 60 | 29.95 | 14.73 | 88.9 | 155850 | 2599 | 3530 | 5 | 706 | 208.4 | 223.1 | 391.0 | 298.4 | 10.20 | 1.23 | 1189.5 | 142.1 | 156.8 | — | 0.53 | 4.5 | 19.2 | 225.2 | 193.3 | 996 |
| 5 | 13 | 60 | 29.98 | 14.73 | 88.9 | 156050 | 2601 | 3784 | 5 | 757 | 207.4 | 222.1 | 390.7 | 295.3 | 10.19 | 1.40 | 1188.1 | 152.6 | 167.3 | — | 0.83 | 4.5 | 19.2 | 225.1 | 193.2 | 994 |
| 6 | 13 | 60 | 29.96 | 14.72 | 88.1 | 156250 | 2604 | 3922 | 5 | 784 | 208.0 | 222.7 | 391.1 | 294.9 | 10.41 | 1.42 | 1187.9 | 159.4 | 174.1 | — | 0.94 | 4.6 | 19.3 | 225.2 | 193.3 | 994 |
| 7 | 15 | 60 | 29.95 | 14.71 | 86.4 | 143500 | 2392 | 2266 | 5 | 453 | 252.2 | 266.9 | 407.6 | 294.1 | 13.19 | 1.89 | 1187.1 | 82.9 | 97.6 | 2.1 | — | 4.6 | 19.3 | 225.8 | 193.9 | 993 |
| 8 | 15 | 60 | 29.94 | 14.71 | 87.4 | 142400 | 2373 | 2518 | 5 | 504 | 250.1 | 264.8 | 406.9 | 290.4 | 13.20 | 2.08 | 1185.3 | 93.6 | 108.3 | — | 0.27 | 4.6 | 19.3 | 225.8 | 193.9 | 991 |
| 9 | 15 | 60 | 29.93 | 14.70 | 87.7 | 144650 | 2411 | 2858 | 5 | 572 | 251.8 | 266.5 | 407.8 | 291.8 | 13.34 | 2.02 | 1186.0 | 110.6 | 125.3 | — | 0.48 | 4.5 | 19.2 | 225.6 | 193.7 | 992 |
| 10 | 15 | 60 | 29.93 | 14.70 | 87.4 | 147950 | 2466 | 3153 | 5 | 631 | 253.7 | 268.4 | 408.1 | 294.8 | 13.23 | 1.85 | 1187.4 | 123.5 | 140.2 | — | 0.55 | 4.5 | 19.2 | 225.5 | 193.6 | 993 |
| 11 | 16 | 60 | 30.10 | 14.79 | 81.9 | 155330 | 2589 | 3739 | 5 | 748 | 251.9 | 266.7 | 406.9 | 297.2 | 13.40 | 1.69 | 1188.5 | 152.4 | 167.2 | — | 0.77 | 4.6 | 19.4 | 225.5 | 193.6 | 994 |
| 12 | 16 | 60 | 30.07 | 14.77 | 85.4 | 156020 | 2600 | 2428 | 4 | 607 | 253.6 | 268.4 | 407.7 | 292.5 | 13.21 | 1.99 | 1186.3 | 118.0 | 132.8 | — | 0.59 | 4.6 | 19.4 | 225.7 | 193.8 | 992 |
| 13 | 16 | 60 | 30.09 | 14.78 | 84.8 | 156180 | 2603 | 2653 | 5 | 531 | 251.1 | 265.9 | 407.1 | 294.2 | 13.19 | 1.86 | 1187.1 | 101.7 | 116.5 | — | 0.22 | 4.7 | 19.5 | 226.0 | 194.1 | 993 |
| 14 | 16 | 60 | 30.06 | 14.77 | 85.7 | 155640 | 2594 | 3183 | 8 | 398 | 252.5 | 267.3 | 406.7 | 294.0 | 13.12 | 1.88 | 1187.0 | 71.4 | 86.2 | 6.2 | — | 4.6 | 19.4 | 225.7 | 193.8 | 993 |
| 15 | 16 | 60 | 30.04 | 14.76 | 87.4 | 159900 | 2565 | 2828 | 5 | 566 | 252.6 | 267.4 | 406.9 | 293.8 | 13.15 | 1.89 | 1186.9 | 109.8 | 124.6 | — | 0.38 | 4.6 | 19.4 | 225.5 | 193.6 | 993 |
| 16 | 17 | 60 | 30.15 | 14.81 | 82.4 | 155680 | 2595 | 2947 | 4 | 737 | 253.3 | 268.1 | 408.0 | 295.4 | 13.42 | 1.83 | 1187.6 | 147.9 | 162.7 | — | 0.84 | 4.6 | 19.4 | 226.0 | 194.1 | 993 |
| 17 | 17 | 60 | 30.16 | 14.81 | 84.9 | 156590 | 2610 | 3400 | 5 | 680 | 251.6 | 266.4 | 407.8 | 297.9 | 13.21 | 1.67 | 1188.9 | 134.2 | 149.0 | — | 0.51 | 4.6 | 19.4 | 226.0 | 194.1 | 994 |
| 18 | 19 | 60 | 29.87 | 14.67 | 88.9 | 155200 | 2587 | 3821 | 8 | 478 | 252.2 | 266.9 | 407.7 | 292.1 | 12.91 | 1.99 | 1186.3 | 88.2 | 102.9 | — | 0.07 | 4.5 | 19.2 | 225.5 | 193.6 | 992 |
| 19 | 19 | 60 | 29.84 | 14.66 | 91.0 | 159050 | 2651 | 3537 | 5 | 707 | 256.6 | 271.3 | 409.6 | 295.2 | 13.34 | 1.87 | 1187.6 | 141.6 | 156.3 | — | 0.76 | 4.6 | 19.3 | 225.3 | 193.4 | 994 |
| 20 | 19 | 60 | 29.82 | 14.65 | 89.5 | 163630 | 2727 | 3748 | 4 | 937 | 252.3 | 267.0 | 409.0 | 298.8 | 13.06 | 1.62 | 1189.4 | 197.2 | 211.9 | — | 1.14 | 4.5 | 19.2 | 225.3 | 193.4 | 996 |
| 21 | 19 | 60 | 29.80 | 14.63 | 89.8 | 172590 | 2877 | 3966 | 5 | 793 | 242.3 | 256.9 | 406.5 | 296.8 | 12.59 | 1.68 | 1188.5 | 159.7 | 174.3 | — | 0.87 | 4.5 | 19.1 | 225.3 | 193.4 | 995 |
| 22 | 19 | 60 | 29.83 | 14.65 | 84.7 | 162920 | 2715 | 4611 | 8 | 576 | 239.9 | 254.6 | 406.0 | 299.1 | 12.13 | 1.55 | 1189.6 | 113.7 | 128.4 | — | 0.13 | 4.6 | 19.3 | 225.3 | 193.4 | 996 |
| 23 | 20 | 60 | 29.78 | 14.63 | 85.7 | 159060 | 2651 | 3360 | 5 | 672 | 252.4 | 267.0 | 409.0 | 299.9 | 13.68 | 1.57 | 1189.8 | 133.4 | 148.0 | — | 0.38 | 0.4 | 15.0 | 214.3 | 182.3 | 1007 |
| 24 | 21 | 60 | 29.68 | 14.58 | 86.0 | 159290 | 2655 | 3798 | 5 | 760 | 251.6 | 266.2 | 408.1 | 300.2 | 14.22 | 1.54 | 1190.0 | 159.9 | 174.5 | — | 0.69 | 8.8 | 23.4 | 235.8 | 204.0 | 986 |
| 25 | 20 | 60 | 29.76 | 14.62 | 92.3 | 154940 | 2582 | 2540 | 5 | 508 | 252.2 | 266.8 | 408.4 | 295.3 | 13.53 | 1.83 | 1187.6 | 96.9 | 111.5 | — | 0.07 | 4.6 | 19.2 | 225.3 | 193.4 | 994 |
| 26 | 21 | 60 | 29.63 | 14.55 | 88.0 | 155170 | 2586 | 2786 | 5 | 557 | 254.3 | 268.9 | 409.9 | 299.9 | 14.56 | 1.59 | 1189.9 | 106.9 | 121.5 | 0.2 | — | 4.7 | 19.3 | 225.3 | 193.4 | 996 |
| 27 | 20 | 60 | 29.78 | 14.63 | 90.9 | 158570 | 2643 | 3302 | 5 | 660 | 254.3 | 268.9 | 409.4 | 295.7 | 13.69 | 1.83 | 1187.8 | 135.7 | 150.3 | — | 0.65 | 4.6 | 19.2 | 225.4 | 193.5 | 994 |
| 28 | 20 | 60 | 29.76 | 14.62 | 92.1 | 160660 | 2678 | 3640 | 5 | 728 | 254.1 | 268.7 | 408.3 | 293.5 | 13.57 | 1.93 | 1186.8 | 151.5 | 166.1 | — | 0.97 | 4.6 | 19.2 | 225.2 | 193.3 | 993 |
| 29 | 21 | 60 | 29.66 | 14.57 | 86.4 | 164820 | 2747 | 4032 | 5 | 806 | 251.0 | 265.6 | 408.3 | 298.2 | 14.29 | 1.66 | 1189.0 | 166.9 | 181.5 | — | 0.89 | 4.6 | 19.2 | 225.0 | 193.1 | 995 |
| 30 | 21 | 60 | 29.67 | 14.58 | 88.0 | 165550 | 2759 | 3714 | 5 | 743 | 188.5 | 203.1 | 385.6 | 302.9 | 9.91 | 0.86 | 1191.7 | 147.9 | 162.5 | — | 0.36 | 4.6 | 19.2 | 224.9 | 193.0 | 996 |

—General Observations and Deduced Data.

| Exhaust Temperature at Turbine, Degrees Fahr. | Heat in Water at Exhaust Temperature above 32°, B.t.u. per Lb. | Heat in Live Steam above Exhaust Temperature, B.t.u. per Lb. | Temperature of Pump Suction, Degrees Fahr. | Outlet Temperature of Thrust Bearing Cooling Water, Degrees Fahr. | Cooling Water used Gallons per Minute. | Density of Water Pumped Lb. per Cu.Ft. | Water Pumped during Run, Cu.Ft. | Water Rate, Cu.Ft. per Minute. | Water Rate, Gallons per Minute. | Water Rate, Lb. per Minute. | Discharge Pressure, Lb. per Sq.In. Gage. | Discharge Pressure, Feet Water. | Suction Pressure, in Mercury. | Suction Pressure, Feet Water. | Vertical Distance between Suction and Discharge Pipes, Feet. | Static Head, Feet Water. | Velocity Head in Discharge Pipe, Feet Water. | Velocity Head in Suction Pipe, Feet Water. | Total Head Pumped against, Feet Water. | Work Done, Foot Pounds per Minute. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 | 36 | 37 | 38 | 39 | 40 | 41 | 42 | 43 | 44 | 45 |
| 2249 | 193.0 | 994.4 | 78.9 | 83.7 | 4.1 | 62.61 | 4669 | 7.78 | 58.2 | 487 | 32.82 | 75.48 | 6.7 | 7.6 | 1.7 | 748.9 | 0.1 | 0.0 | 749.0 | 364900 |
| 2245 | 192.6 | 995.6 | 79.3 | 84.9 | 4.4 | 62.60 | 9942 | 16.57 | 124.0 | 1037 | 331.8 | 763.2 | 2.2 | 2.5 | - | 762.4 | 0.5 | 0.2 | 762.7 | 791100 |
| 2252 | 193.3 | 996.1 | 79.0 | 82.8 | 4.7 | 62.56 | 15376 | 25.63 | 191.7 | 1603 | 348.3 | 732.7 | 4.9 | 5.5 | - | 728.9 | 1.1 | 0.4 | 729.6 | 1169700 |
| 2252 | 193.3 | 996.2 | 80.0 | 83.7 | 4.5 | 62.57 | 19716 | 32.86 | 245.8 | 2056 | 302.9 | 697.1 | 2.6 | 2.9 | - | 695.9 | 1.8 | 0.6 | 697.1 | 1433300 |
| 2251 | 193.2 | 994.9 | 81.0 | 84.3 | 4.2 | 62.54 | 24778 | 41.30 | 308.9 | 2583 | 275.4 | 634.1 | 1.0 | 1.1 | - | 634.7 | 2.8 | 0.9 | 636.6 | 1644100 |
| 2252 | 193.3 | 994.6 | 80.7 | 83.6 | 4.5 | 62.57 | 28269 | 47.12 | 352.5 | 2948 | 233.8 | 538.1 | -1.5 | -1.7 | - | 641.5 | 3.6 | 1.2 | 643.9 | 1898200 |
| 2258 | 193.9 | 993.2 | 81.1 | 84.7 | 3.2 | 62.54 | 4952 | 8.25 | 61.8 | 516 | 274.0 | 630.9 | 8.3 | 9.4 | - | 623.2 | 0.1 | 0.0 | 623.3 | 321700 |
| 2258 | 193.9 | 991.4 | 81.0 | 84.8 | 3.0 | 62.56 | 9290 | 15.48 | 115.8 | 969 | 272.6 | 627.5 | 6.5 | 7.3 | - | 621.9 | 0.4 | 0.1 | 622.2 | 602.700 |
| 2256 | 193.7 | 992.3 | 80.9 | 85.0 | 3.1 | 62.57 | 14256 | 23.76 | 177.8 | 1487 | 272.0 | 626.0 | 5.5 | 6.2 | - | 621.5 | 0.9 | 0.3 | 622.1 | 924900 |
| 2255 | 193.6 | 993.8 | 81.0 | 83.9 | 5.6 | 62.56 | 18065 | 30.11 | 225.2 | 1884 | 274.2 | 631.2 | 3.2 | 3.6 | - | 629.3 | 1.5 | 0.5 | 630.3 | 1187200 |
| 2255 | 193.6 | 994.9 | 79.1 | 82.5 | 3.3 | 62.59 | 24664 | 41.11 | 307.5 | 2573 | 273.9 | 630.2 | -1.4 | -1.6 | - | 633.5 | 2.8 | 0.9 | 635.4 | 1634800 |
| 2257 | 193.8 | 992.5 | 79.0 | 86.4 | 3.4 | 62.60 | 4798 | 8.00 | 59.8 | 501 | 321.2 | 738.9 | 9.0 | 10.2 | - | 730.4 | 0.1 | 0.0 | 730.5 | 365700 |
| 2260 | 194.1 | 993.0 | 78.8 | 85.3 | 3.4 | 62.59 | 4835 | 8.06 | 60.3 | 504 | 321.6 | 739.9 | 8.8 | 9.9 | - | 731.7 | 0.1 | 0.0 | 731.8 | 369100 |
| 2257 | 193.8 | 993.2 | 79.7 | 87.1 | 3.4 | 62.60 | 4751 | 7.92 | 59.2 | 496 | 319.8 | 735.6 | 8.8 | 9.9 | - | 727.4 | 0.1 | 0.0 | 727.5 | 360600 |
| 2255 | 193.6 | 993.3 | 80.3 | 86.2 | 3.4 | 62.58 | 9620 | 16.03 | 120.0 | 1003 | 319.9 | 736.1 | 6.1 | 6.9 | - | 730.9 | 0.4 | 0.1 | 731.2 | 733700 |
| 2260 | 194.1 | 993.5 | 79.4 | 82.3 | 4.8 | 62.60 | 14263 | 23.77 | 177.8 | 1488 | 319.0 | 733.8 | 5.7 | 6.4 | - | 729.1 | 0.9 | 0.3 | 729.7 | 1085900 |
| 2260 | 194.1 | 994.8 | 80.0 | 82.0 | 5.0 | 62.57 | 14320 | 23.87 | 178.6 | 1493 | 318.6 | 733.2 | 5.5 | 6.2 | - | 728.7 | 0.9 | 0.3 | 729.3 | 1089100 |
| 2255 | 193.6 | 992.7 | 80.4 | 83.2 | 3.8 | 62.59 | 14300 | 23.83 | 178.3 | 1492 | 316.7 | 728.6 | 5.5 | 6.2 | - | 724.1 | 0.9 | 0.3 | 724.7 | 1081000 |
| 2253 | 193.4 | 994.2 | 81.3 | 84.8 | 3.6 | 62.60 | 19142 | 31.90 | 238.7 | 1997 | 318.3 | 732.2 | 2.8 | 3.2 | - | 730.7 | 1.7 | 0.6 | 731.8 | 1461600 |
| 2253 | 193.4 | 996.0 | 82.7 | 86.4 | 3.4 | 62.56 | 23778 | 39.63 | 296.5 | 2479 | 316.5 | 728.5 | -0.5 | -0.6 | - | 730.8 | 2.6 | 0.9 | 732.5 | 1816100 |
| 2253 | 193.4 | 995.1 | 82.4 | 86.4 | 2.8 | 62.50 | 24162 | 40.27 | 301.3 | 2517 | 308.8 | 711.5 | -0.9 | -1.0 | - | 714.2 | 2.7 | 0.9 | 716.0 | 1802.100 |
| 2253 | 193.4 | 996.2 | 82.5 | 86.6 | 2.9 | 62.51 | 23512 | 39.19 | 293.2 | 2450 | 315.5 | 726.8 | -0.6 | -0.7 | - | 729.2 | 2.5 | 0.8 | 730.9 | 1790300 |
| 2143 | 182.3 | 1007.5 | 98.3 | 82.0 | 3.6 | 62.52 | 19375 | 32.29 | 241.6 | 2019 | 315.3 | 726.2 | 3.6 | 4.1 | - | 723.8 | 1.7 | 0.6 | 724.9 | 1463500 |
| 2358 | 204.0 | 986.0 | 159.5 | 83.6 | 3.0 | 61.39 | 19626 | 32.71 | 244.7 | 2008 | 309.9 | 726.9 | 3.6 | 4.1 | - | 724.5 | 1.8 | 0.6 | 725.7 | 1457300 |
| 2253 | 193.4 | 994.2 | 167.4 | 87.5 | 3.6 | 61.35 | 5297 | 8.83 | 66.0 | 542 | 313.2 | 735.1 | 8.0 | 9.2 | - | 727.6 | 0.1 | 0.0 | 727.7 | 394100 |
| 2253 | 193.4 | 996.5 | 169.0 | 90.5 | 3.2 | 61.28 | 9242 | 15.40 | 115.2 | 944 | 308.5 | 724.9 | 5.8 | 6.7 | - | 719.9 | 0.4 | 0.1 | 720.2 | 679900 |
| 2254 | 193.5 | 994.3 | 156.5 | 83.5 | 3.5 | 61.47 | 14768 | 24.61 | 184.1 | 1513 | 323.1 | 756.9 | 6.6 | 7.6 | - | 751.0 | 1.0 | 0.3 | 751.7 | 1137300 |
| 2252 | 193.3 | 993.5 | 169.0 | 83.9 | 3.6 | 61.23 | 19679 | 32.80 | 245.4 | 2008 | 315.5 | 742.0 | 3.4 | 3.9 | - | 739.8 | 1.8 | 0.6 | 741.0 | 1488100 |
| 2250 | 193.1 | 995.9 | 170.4 | 84.9 | 3.2 | 61.26 | 24684 | 41.14 | 307.8 | 2520 | 308.4 | 724.9 | -1.0 | -1.2 | - | 727.8 | 2.8 | 0.9 | 729.7 | 1839400 |
| 2249 | 193.0 | 998.7 | 125.7 | 89.0 | 2.5 | 62.09 | 19447 | 32.41 | 242.5 | 2012 | 348.2 | 789.0 | 3.2 | 3.6 | - | 787.1 | 1.7 | 0.6 | 788.2 | 1586300 |

## Table I.—General Observations and Deduced Data.

| | Heat in Steam above 32° F. B.T.U. per Lb. | Steam Pressure at Nozzles, Lb. per Sq. In. Gage. | Steam Pressure at Nozzles, Lb. per Sq. In. Absolute. | Superheat at Nozzle, Degrees Fahr. | Moisture at Nozzle, per Cent. | Exhaust Pressure at Turbine, Lb. per Sq. In. Gage. | Exhaust Pressure at Turbine, Lb. per Sq. In. Absolute. | Exhaust Temperature at Turbine, Degrees Fahr. | Heat in Water at Exhaust Temperature above 32°, B.T.U. per Lb. | Heat in Live Steam above Exhaust Temperature, B.T.U. per Lb. | Temperature of Pump Suction, Degrees Fahr. | Outlet Temperature of Thrust Bearing Cooling Water, Degrees Fahr. | Cooling Water Used Gallons per Minute. | Density of Water Pumped Lb. per Cu. Ft. | Water Pumped during Run, Cu. Ft. | Water Rate, Cu. Ft. per Minute. | Water Rate, Gallons per Minute. | Water Rate, Lb. per Minute. | Discharge Pressure, Lb. per Sq. In. Gage. | Discharge Pressure, Feet Water. | Suction Pressure, in Mercury. | Suction Pressure, Feet Water. | Vertical Distance between Suction and Discharge Pipes, Feet. | Static Head, Feet Water. | Velocity Head in Discharge Pipe, Feet Water. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 | 36 | 37 | 38 | 39 | 40 | 41 | 42 |
| 9 | 1187.4 | 98.6 | 113.3 | — | 0.12 | 4.4 | 19.1 | 224.9 | 193.0 | 994.4 | 78.9 | 83.7 | 4.1 | 62.61 | 466.9 | 7.78 | 58.2 | 487 | 328.2 | 754.8 | 6.7 | 7.6 | 17 | 748.9 | 0.1 |
| 9 | 1188.2 | 113.0 | 127.7 | — | 0.27 | 4.4 | 19.1 | 224.5 | 192.6 | 995.6 | 79.3 | 84.9 | 4.4 | 62.60 | 994.2 | 16.57 | 124.0 | 1037 | 331.8 | 763.2 | 2.2 | 2.5 | • | 762.4 | 0.5 |
| 2 | 1189.4 | 131.6 | 146.3 | — | 0.42 | 4.4 | 19.1 | 225.2 | 193.3 | 996.1 | 79.0 | 82.8 | 4.7 | 62.56 | 1537.6 | 25.63 | 191.7 | 1603 | 328.3 | 732.7 | 4.9 | 5.5 | • | 728.9 | 1.1 |
| 8 | 1189.5 | 142.1 | 156.8 | — | 0.53 | 4.5 | 19.2 | 225.2 | 193.3 | 996.2 | 80.0 | 83.7 | 4.5 | 62.57 | 1971.6 | 32.86 | 245.8 | 2056 | 302.9 | 697.1 | 2.6 | 2.9 | • | 695.9 | 1.8 |
| 0 | 1188.1 | 152.6 | 167.3 | — | 0.83 | 4.5 | 19.2 | 225.1 | 193.2 | 994.9 | 81.0 | 84.3 | 4.2 | 62.54 | 2471.8 | 41.30 | 308.9 | 2583 | 275.4 | 634.1 | 1.0 | 1.1 | • | 634.7 | 2.8 |
| 2 | 1187.9 | 159.4 | 174.1 | — | 0.94 | 4.6 | 19.3 | 225.2 | 193.3 | 994.6 | 80.7 | 83.6 | 4.5 | 62.57 | 2826.9 | 47.12 | 352.5 | 2948 | 233.8 | 538.1 | -1.5 | -1.7 | • | 541.5 | 3.6 |
| 9 | 1187.1 | 82.9 | 97.6 | 2.1 | — | 4.6 | 19.3 | 225.8 | 193.9 | 993.2 | 81.1 | 84.7 | 3.2 | 62.54 | 495.2 | 8.25 | 61.8 | 516 | 274.0 | 630.9 | 8.3 | 9.4 | • | 623.2 | 0.1 |
| 8 | 1185.3 | 93.6 | 108.3 | — | 0.27 | 4.6 | 19.3 | 225.8 | 193.9 | 991.4 | 81.0 | 84.8 | 3.0 | 62.56 | 929.0 | 15.48 | 115.8 | 969 | 272.6 | 627.5 | 6.5 | 7.3 | • | 621.9 | 0.4 |
| 2 | 1186.0 | 110.6 | 125.3 | — | 0.48 | 4.5 | 19.2 | 225.6 | 193.7 | 992.3 | 80.9 | 85.0 | 3.1 | 62.57 | 1425.6 | 23.76 | 177.8 | 1487 | 272.0 | 626.0 | 5.5 | 6.2 | • | 621.5 | 0.9 |
| 5 | 1187.4 | 125.5 | 140.2 | — | 0.55 | 4.5 | 19.2 | 225.5 | 193.6 | 993.8 | 81.0 | 83.9 | 5.6 | 62.56 | 1806.5 | 30.11 | 225.2 | 1884 | 274.2 | 631.2 | 3.2 | 3.6 | • | 629.3 | 1.5 |
| 9 | 1188.5 | 152.4 | 167.2 | — | 0.77 | 4.6 | 19.4 | 225.5 | 193.6 | 994.9 | 79.1 | 82.5 | 3.3 | 62.59 | 2466.4 | 41.11 | 307.5 | 2573 | 273.9 | 630.2 | -1.4 | -1.6 | • | 633.5 | 2.8 |
| 9 | 1186.3 | 118.0 | 132.8 | — | 0.59 | 4.6 | 19.4 | 225.7 | 193.8 | 992.5 | 79.0 | 86.4 | 3.4 | 62.60 | 479.8 | 8.00 | 59.8 | 501 | 321.2 | 738.9 | 9.0 | 10.2 | • | 730.4 | 0.1 |
| 6 | 1187.1 | 101.7 | 116.5 | — | 0.22 | 4.7 | 19.5 | 226.0 | 194.1 | 993.0 | 78.8 | 85.3 | 3.4 | 62.59 | 483.5 | 8.06 | 60.3 | 504 | 321.6 | 739.9 | 8.8 | 9.9 | • | 731.7 | 0.1 |
| 8 | 1187.0 | 71.4 | 86.2 | 6.2 | — | 4.6 | 19.4 | 225.7 | 193.8 | 993.2 | 79.7 | 87.1 | 3.4 | 62.60 | 475.1 | 7.92 | 59.2 | 496 | 319.8 | 735.6 | 8.8 | 9.9 | • | 727.4 | 0.1 |
| 9 | 1186.9 | 109.8 | 124.6 | — | 0.38 | 4.6 | 19.4 | 225.5 | 193.6 | 993.3 | 80.3 | 86.2 | 3.4 | 62.58 | 962.0 | 16.03 | 120.0 | 1003 | 319.9 | 736.1 | 6.1 | 6.9 | • | 730.9 | 0.4 |
| 3 | 1187.6 | 147.9 | 162.7 | — | 0.84 | 4.6 | 19.4 | 226.0 | 194.1 | 993.5 | 79.4 | 82.3 | 4.8 | 62.60 | 1426.3 | 23.77 | 177.8 | 1488 | 319.0 | 733.8 | 5.7 | 6.4 | • | 729.1 | 0.9 |
| 7 | 1188.9 | 134.2 | 149.0 | — | 0.51 | 4.6 | 19.4 | 226.0 | 194.1 | 994.8 | 80.0 | 82.0 | 5.0 | 62.57 | 1432.0 | 23.87 | 178.6 | 1493 | 318.6 | 733.2 | 5.5 | 6.2 | • | 728.7 | 0.9 |
| 9 | 1186.3 | 88.2 | 102.9 | — | 0.07 | 4.5 | 19.2 | 225.5 | 193.6 | 992.7 | 80.4 | 83.2 | 3.8 | 62.59 | 1430.0 | 23.83 | 178.3 | 1492 | 316.7 | 728.6 | 5.5 | 6.2 | • | 724.1 | 0.9 |
| 7 | 1187.6 | 141.6 | 156.3 | — | 0.76 | 4.6 | 19.3 | 225.3 | 193.4 | 994.2 | 81.3 | 84.8 | 3.6 | 62.60 | 1914.2 | 31.90 | 238.7 | 1997 | 318.3 | 732.2 | 2.8 | 3.2 | • | 730.7 | 1.7 |
| 2 | 1189.4 | 197.2 | 211.9 | — | 1.14 | 4.5 | 19.2 | 225.3 | 193.4 | 996.0 | 82.7 | 86.4 | 3.4 | 62.56 | 2377.8 | 39.63 | 296.5 | 2479 | 316.5 | 728.5 | -0.5 | -0.6 | • | 730.8 | 2.6 |
| 8 | 1188.5 | 159.7 | 174.3 | — | 0.87 | 4.5 | 19.1 | 225.3 | 193.4 | 995.1 | 82.4 | 86.4 | 2.8 | 62.50 | 2416.2 | 40.27 | 301.3 | 2517 | 308.8 | 711.5 | -0.9 | -1.0 | • | 711.2 | 2.7 |
| 5 | 1189.6 | 113.7 | 128.4 | — | 0.13 | 4.6 | 19.3 | 225.3 | 193.4 | 996.2 | 82.5 | 86.6 | 2.9 | 62.51 | 2351.2 | 39.19 | 293.2 | 2450 | 315.5 | 726.8 | -0.6 | -0.7 | • | 729.2 | 2.5 |
| 7 | 1189.8 | 133.4 | 148.0 | — | 0.38 | 0.4 | 15.0 | 214.3 | 182.3 | 1007.5 | 98.3 | 82.0 | 3.6 | 62.52 | 1937.5 | 32.29 | 241.6 | 2019 | 315.3 | 726.2 | 3.6 | 4.1 | • | 723.8 | 1.7 |
| 4 | 1190.0 | 159.9 | 174.5 | — | 0.69 | 8.8 | 23.4 | 235.8 | 204.0 | 986.0 | 159.5 | 83.6 | 3.0 | 61.39 | 1962.6 | 32.71 | 244.7 | 2008 | 309.9 | 726.9 | 3.6 | 4.1 | • | 724.5 | 1.8 |
| 3 | 1187.6 | 96.9 | 111.5 | — | 0.07 | 4.6 | 19.2 | 225.3 | 193.4 | 994.2 | 167.4 | 87.5 | 3.6 | 61.35 | 529.7 | 8.83 | 66.0 | 542 | 313.2 | 735.1 | 8.0 | 9.2 | • | 727.6 | 0.2 |
| 9 | 1189.9 | 106.9 | 121.5 | 0.2 | — | 4.7 | 19.3 | 225.3 | 193.4 | 996.5 | 169.0 | 90.5 | 3.2 | 61.28 | 924.2 | 15.40 | 115.2 | 944 | 308.5 | 724.9 | 5.8 | 6.7 | • | 719.9 | 0.4 |
| 3 | 1187.8 | 135.7 | 150.3 | — | 0.65 | 4.6 | 19.2 | 225.4 | 193.5 | 994.3 | 156.5 | 83.5 | 3.5 | 61.47 | 1476.8 | 24.61 | 184.1 | 1513 | 323.1 | 756.9 | 6.6 | 7.6 | • | 751.0 | 1.0 |
| 3 | 1186.8 | 151.5 | 166.1 | — | 0.97 | 4.6 | 19.2 | 225.2 | 193.3 | 993.5 | 169.0 | 83.9 | 3.6 | 61.23 | 1967.9 | 32.80 | 245.4 | 2008 | 315.5 | 742.0 | 3.4 | 3.9 | • | 739.8 | 1.1 |
| 6 | 1189.0 | 166.9 | 181.5 | — | 0.89 | 4.6 | 19.2 | 225.0 | 193.1 | 995.9 | 170.4 | 84.9 | 3.2 | 61.26 | 2468.4 | 41.14 | 307.8 | 2520 | 308.4 | 724.9 | -1.0 | -1.2 | • | 727.8 | 2.1 |
| 6 | 1191.7 | 147.9 | 162.5 | — | 0.36 | 4.6 | 19.2 | 224.9 | 193.0 | 998.7 | 125.7 | 89.0 | 2.5 | 62.09 | 1944.7 | 32.41 | 242.5 | 2012 | 340.2 | 789.0 | 3.2 | 3.6 | • | 787.1 | 1.9 |

| SUCTION AND DISCHARGE PIPES, FEET. | STATIC HEAD, FEET WATER | VELOCITY HEAD IN DISCHARGE PIPE, FEET WATER. | VELOCITY HEAD IN SUCTION PIPE, FEET WATER. | TOTAL HEAD PUMPED AGAINST, FEET WATER. | WORK DONE, FOOT-POUNDS PER MINUTE. | WATER HORSE POWER | STEAM ECONOMY PER WATER HORSEPOWER, LB PER HOUR. | HEAT ECONOMY PER WATER HORSEPOWER, B.T.U. PER MINUTE ABOVE 32° F. | HEAT ECONOMY PER WATER HORSEPOWER, B.T.U. PER MINUTE ABOVE EXHAUST TEMPERATURE. | DUTY, FOOT-POUNDS PER 1 000 000 B.T.U. ABOVE 32° F. | DUTY, FOOT-POUNDS PER 1000 000 B.T.U. ABOVE EXHAUST STEAM TEMPERATURE. | STEAM PER 1000 LB. WATER PUMPED, LB. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 41 | 42 | 43 | 44 | 45 | 46 | 47 | 48 | 49 | 50 | 51 | 52 |
| 7 | 748.9 | 0.1 | 0.0 | 749.0 | 364900 | 11.06 | 234.7 | 4646 | 3890 | 7 103 000 | 8489000 | 88.81 |
| " | 762.4 | 0.5 | 0.2 | 762.7 | 791100 | 23.97 | 123.9 | 2455 | 2057 | 13 442 000 | 16 043 000 | 47.74 |
| " | 728.9 | 1.1 | 0.4 | 729.6 | 1169700 | 35.45 | 93.7 | 1859 | 1555 | 17 756 000 | 21 222 000 | 34.52 |
| " | 695.9 | 1.8 | 0.6 | 697.1 | 1433300 | 43.43 | 81.3 | 1611 | 1349 | 20 480 000 | 24 463 000 | 28.61 |
| " | 634.7 | 2.8 | 0.9 | 636.6 | 1644100 | 49.82 | 76.0 | 1504 | 1260 | 21 940 000 | 26 190 000 | 24.42 |
| " | 641.5 | 3.6 | 1.2 | 643.9 | 1898200 | 57.52 | 68.2 | 1350 | 1130 | 24 444 000 | 29 204 000 | 22.17 |
| " | 623.2 | 0.1 | 0.0 | 623.3 | 321700 | 9.75 | 232.4 | 4699 | 3848 | 7 176 000 | 8576000 | 73.17 |
| " | 621.9 | 0.4 | 0.1 | 622.2 | 602.700 | 18.26 | 137.9 | 2724 | 2279 | 12 113 000 | 14480000 | 43.33 |
| " | 621.5 | 0.9 | 0.3 | 622.1 | 924900 | 28.03 | 102.0 | 2015 | 1686 | 16 375 000 | 19573000 | 32.04 |
| " | 629.3 | 1.5 | 0.5 | 630.3 | 1187200 | 35.98 | 87.6 | 1734 | 1452 | 19 029 000 | 22 727 000 | 27.90 |
| " | 633.5 | 2.8 | 0.9 | 635.4 | 1634800 | 49.54 | 75.5 | 1495 | 1252 | 22 072 000 | 26 358000 | 24.22 |
| " | 730.4 | 0.1 | 0.0 | 730.5 | 365700 | 11.08 | 219.1 | 4333 | 3625 | 7616000 | 9103000 | 80.84 |
| " | 731.7 | 0.1 | 0.0 | 731.8 | 369100 | 11.18 | 237.3 | 4695 | 3928 | 7029000 | 8401000 | 87.67 |
| " | 727.4 | 0.1 | 0.0 | 727.5 | 360600 | 10.93 | 291.2 | 5761 | 4821 | 5 728000 | 6845000 | 107.02 |
| " | 730.9 | 0.4 | 0.1 | 731.2 | 733700 | 22.23 | 127.2 | 2516 | 2106 | 13 116 000 | 15670000 | 46.97 |
| " | 729.1 | 0.9 | 0.3 | 729.7 | 1085900 | 32.91 | 89.5 | 1773 | 1483 | 18613000 | 22 252000 | 33.01 |
| " | 728.7 | 0.9 | 0.3 | 729.3 | 1089100 | 33.00 | 103.0 | 2042 | 1708 | 16 161 000 | 19321000 | 37.95 |
| " | 724.1 | 0.9 | 0.3 | 724.7 | 1081000 | 32.76 | 116.6 | 2306 | 1930 | 14 310 000 | 17098000 | 42.69 |
| " | 730.7 | 1.7 | 0.6 | 733.8 | 1461600 | 44.29 | 79.9 | 1581 | 1323 | 20 873 000 | 24943000 | 29.52 |
| " | 730.8 | 2.6 | 0.9 | 732.5 | 1816100 | 55.03 | 68.1 | 1350 | 1131 | 24 444 000 | 29178000 | 25.29 |
| " | 714.2 | 2.7 | 0.9 | 716.0 | 1802.100 | 54.61 | 72.6 | 1439 | 1205 | 22 933 000 | 27386000 | 26.26 |
| " | 729.2 | 2.5 | 0.8 | 730.9 | 1790300 | 54.25 | 85.0 | 1685 | 1411 | 19585 000 | 23388000 | 31.37 |
| " | 723.8 | 1.7 | 0.6 | 724.9 | 1463500 | 44.35 | 75.8 | 1502 | 1272 | 21971 000 | 25943000 | 27.74 |
| " | 724.5 | 1.8 | 0.6 | 725.7 | 1457300 | 44.16 | 86.0 | 1706 | 1413 | 19343000 | 23355000 | 31.52 |
| " | 727.6 | 0.1 | 0.0 | 727.7 | 394100 | 11.94 | 212.7 | 4210 | 3525 | 7838000 | 9362.000 | 78.16 |
| " | 719.9 | 0.4 | 0.1 | 720.2 | 679900 | 20.60 | 135.2 | 2682 | 2246 | 12 304.000 | 14693000 | 49.18 |
| " | 751.0 | 1.0 | 0.3 | 751.7 | 1137300 | 34.46 | 95.8 | 1897 | 1588 | 17596000 | 20781 000 | 36.37 |
| " | 739.8 | 1.8 | 0.6 | 741.0 | 1488100 | 45.09 | 80.7 | 1597 | 1337 | 20664000 | 24682000 | 30.21 |
| " | 727.8 | 2.8 | 0.9 | 729.7 | 1839000 | 55.73 | 72.3 | 1434 | 1201 | 23013000 | 27477000 | 26.66 |
| " | 787.1 | 1.7 | 0.6 | 788.2 | 1586300 | 48.07 | 77.3 | 1535 | 1286 | 21 498000 | 25661000 | 30.76 |

inclusive, with warm water. Run 30 was made with low steam pressure as well as high water-discharge pressure.

The source of each column in Table I is as follows:

Column 1. For reference.

" 2. From data taken during test.

" 3. From data taken during test.

" 4. From data taken during each run corrected to 32 degrees F, by Smithsonian Meteorological Tables.

" 5. Column 4 multiplied by 0.4912 pound per square inch, equivalent to one inch mercury at 32 degrees F.

" 6. From data taken during each run.

" 7. From difference of speed-counter readings at beginning and end of each run.

" 8. Column 7 divided by column 3.

" 9. From weighings made during each run corrected for calibration error.

" 10. From data taken during each run.

" 11. Column 9 divided by column 10.

" 12. From data taken each run corrected for calibration error.

" 13. Column 12 plus column 5.

" 14. From data taken during each run corrected for calibration error and for exposed stem.

" 15. From data taken during each run corrected for calibration error and for exposed stem.

" 16. From data taken during each run corrected for water above mercury.

" 17. From formula

$$y = 100\left(1 - \frac{H + k(t_1 - t) - q}{r}\right);$$

in which $y$ is the percentage of moisture, $r$ is the latent heat, $q$ is the heat of the liquid corresponding to temperature in column 14, $H$ is the total heat, $t$ the saturated steam temperature corresponding to the pressure in column 16, $t_1$

is the temperature in column 15, and $k$ is the proper value of the specific heat of superheated steam.

Column 18. From formula

$$H_1 = q + \frac{100 - y}{100} r;$$

in which $q$ is the heat of the liquid, $r$ is the latent heat corresponding to temperature in column 14, and $y$ is the percentage of moisture in column 17.

" 19. From data taken during each run, corrected for calibration error.

" 20. Column 19 plus column 5.

" 21. From formula

$$t_s = \frac{H_1 - H}{k};$$

in which $H_1$ is the value in column 18, $H$ is the total heat of steam at pressure in column 20, and $k$ is the proper value of the specific heat of superheated steam.

" 22. From formula

$$y = 100 \left(1 - \frac{H_1 - q}{r}\right);$$

in which $H_1$ is the value in column 18, $q$ is the heat of the liquid, and $r$ is the latent heat corresponding to the steam pressure in column 20.

" 23. From data taken during each run, corrected for calibration error.

" 24. Column 23 plus column 5.

" 25. From data taken during each run, corrected for calibration error and for exposed stem.

" 26. From steam tables, corresponding to temperature in column 25.

" 27. Column 18 minus column 26.

" 28. From data taken during each run.

Column 29. From data taken during each run.

" 30. From data taken during each run, corrected for calibration error.

" 31. From test of sample taken each run.

" 32. From fall in level of suction tank, recently calibrated, and venturi-meter readings taken during each run. For runs in which the suction-tank level was allowed to fall during a portion of the time only, the amount of water pumped was determined by multiplying the cubic feet corresponding to the drop in level by the ratio of the venturi-meter readings and the ratio of the times for the whole run to that portion during which the fall in water level took place. For the runs in which no drop in water level occurred the amount of water pumped was obtained from the venturi-meter readings corrected by corresponding suction-tank readings in other runs, and for temperature and density of water.

" 33. Column 32 divided by column 3.

" 34. Column 33 multiplied by 7.481 gallons per cubic foot.

" 35. Column 33 multiplied by column 31.

" 36. From data taken during each run, corrected for calibration error and reduced to center line of discharge pipe.

" 37. Column 36 multiplied by 144, and divided by column 31.

" 38. From data taken during each run, reduced to center line of suction pipe.

" 39. Column 38 multiplied by 0.4905 pound per square inch for one inch mercury, multiplied by 144 and divided by column 31.

" 40. From measurement after test.

" 41. Column 37 plus column 39 plus column 40.

Column 42. From formula

$$h_1 = \frac{v_1^2}{2g};$$

in which $v_1$, the velocity in the discharge pipe in feet per second, is equal to column 33 divided by 60 and by $\frac{7.383}{144}$ square feet area of standard 3-inch wrought-iron pipe; and $g$ is 32.15 feet per second per second.

" 43. From formula

$$h_2 = \frac{v_2^2}{2g};$$

in which $v_2$, the velocity in the suction pipe in feet per second, is equal to column 33 divided by 60 and by $\frac{12.730}{144}$ square feet area of standard 4-inch wrought-iron pipe; and $g$ is 32.15 feet per second per second.

" 44. Column 41 plus column 42 minus column 43.

" 45. Column 35 multiplied by column 44.

" 46. Column 45 divided by 33,000 foot-pounds per minute per horsepower.

" 47. Column 9 divided by column 46.

" 48. Column 9 multiplied by column 18 and divided by column 3 and by column 46.

" 49. Column 9 multiplied by column 27 and divided by column 3 and by column 46.

" 50. 33,000 foot-pounds per minute per horsepower divided by column 48 and multiplied by 1,000,000.

" 51. 33,000 foot-pounds per minute per horsepower divided by column 49 and multiplied by 1,000,000.

" 52. Column 9 divided by column 3 and by column 35, and then multiplied by 1,000.

The quality of the steam in Table I is given in terms of the percentage of moisture, column 17, rather than the dryness fraction, because the former is directly used in reducing the performance of a turbine to standard conditions for compari-

son with other tests. The moisture or superheat at the nozzles, columns 21 and 22, has been found to show that all the steam used—not simply the dry steam—must be considered in economy calculations, columns 47 and 52.

Heat economy and duty have been based upon the heat in the steam above water at exhaust temperature and above 32 degrees F. The former is the rational method, since the pump is not debited with the inefficiency of any other apparatus. If any fixed temperature is chosen, however, the starting temperature of all steam tables is naturally the best choice.

## SUMMARY AND CONCLUSIONS.

With a normal speed of 2,600 revolutions per minute, as marked on the steam turbine, the capacity of the centrifugal pump may be obtained from the data of the first six runs, Table I. The discharge pressure was maintained above 300 pounds per square inch gage up to a water rate of 246 gallons per minute. At a water rate of 309 gallons per minute the discharge pressure dropped to 275 pounds per square inch gage. Therefore, the pump may be rated as having a full-load capacity of 250 gallons per minute or 125,000 pounds per hour.

If we also require a discharge pressure from 50 to 60 pounds per square inch in excess of the boiler pressure, the pump may be rated as suitable for a boiler pressure of 250 pounds per square inch gage.

A steam pressure of 250 pounds per square inch gage was therefore selected for the remaining tests, and runs were made with water rates of one-quarter, one-half, three-quarters, one, and one and one-quarter times the full-load water rate of 250 gallons per minute.

The results of the test as given in Table I have been plotted as curves A, B, C and D, in order to bring out the results of the various possible methods of operating the machine.

The effect of varying the number of nozzles open is evident from curves A. The discharge pressure, and consequently the total head pumped against, is maintained practically constant

by the Foster pump governor, the speed increasing as shown with an increase of the water rate. Against a constant head the water horsepower is directly proportional to the rate of pumping water. As the number of nozzles open is decreased from 8 to 5, or to 4, the steam pressure at the nozzles is increased, resulting in a more economical performance of the machine. Thus, the steam used per hour, the steam per 1,000 pounds of water pumped, the steam per water horsepower and the heat economy are all reduced, and the duty is increased.

The above result suggests that in the operation of the machine a pressure gage should be attached to the steam ring to show the nozzle pressure, which should always be maintained as near the supply pressure as possible by closing nozzles when the load decreases. Of course, if nozzles are not opened again as the load increases the speed and consequently the discharge pressure will not be maintained at the required value.

In order to show what might be expected from this machine if it were provided with at least two additional nozzle valves and operated with a mean steam pressure at the nozzles of 225 pounds per square inch gage by adjusting the number of nozzles open according to the load, the lower two groups of curves have been plotted on drawing marked " Curves D." The various points marked on the left-hand diagram have been obtained from the curves on drawing marked " Curves A" by taking as abscissae the steam pressure at nozzles corresponding to 100, 200 and 300 gallons per minute, respectively, and by taking as ordinates the steam used per hour, steam economy, etc., corresponding to these same water rates. Hence, each curve shows the change in the quantity it represents as the steam pressure at nozzles is varied by changing the number of nozzles open while the water rate and the head are maintained constant. The several ordinates at a nozzle pressure of 225 pounds per square inch gage have been plotted on the right-hand diagram in terms of water rate. Although it was necessary to greatly prolong some of the curves beyond the points plotted in the left-hand diagram, it is believed that a

fair degree of accuracy has been secured by carefully comparing curves in the same group.

The effect upon the economical working of the machine of pumping against different heads of water may be seen by curves B. To pump against a lower head a lower speed is sufficient and also a lower steam pressure at the nozzles, which results in less steam used per hour and less steam per 1,000 pounds of water pumped. Due to the lower head pumped against, however, the water horsepower is decreased to a greater extent than the steam used. Consequently, the steam economy in pounds per hour per water horsepower and the heat economy in B.t.u.'s per minute per water horsepower are both increased, and the duty is decreased.

If the object of the feed pump were to do as much work as possible in the most economical manner, then within the range tested it would be desirable to pump against a high discharge pressure. But, as the difference between the discharge pressure at the pump and the pressure at the boiler is only used up by wasteful resistances, it is most economical to run the machine at the lowest possible discharge pressure that will force a sufficient quantity of water into the boilers.

While the pump should be capable of producing continuously at normal speed a pressure about 60 pounds per square inch in excess of the boiler pressure, the attempt should be made to operate the machine with an excess of only 30 pounds per square inch, for less steam will then be used to pump the required amount of feed water. As an excessive resistance might be temporarily encountered in the feed line or heater, the machine should be capable of operating at a higher speed against a pressure 90 pounds per square inch in excess of the boiler pressure. The pump tested meets these requirements for a boiler pressure of 250 pounds per square inch gage.

A great advantage of the centrifugal pump is the steadiness of this discharge pressure; but within the capacity of the pump this discharge pressure depends upon the speed of the pump rather than upon the resistance encountered in the discharge pipe. The maximum discharge pressure is determined

by the possible governor adjustment and by the safe speed of the machine. For the pump tested this maximum is about 340 pounds per square inch gage at full load.

The effect of the method of governing the turbine—either by the turbine governor at a constant speed or by the Foster pump governor at a constant discharge pressure—is shown in curves C. When the speed is maintained constant the discharge pressure at first increases slightly and then decreases with an increase in the water rate. When the discharge pressure is maintained constant the speed at first slightly decreases and then increases with an increase in the water rate. The greatest difference in the economy curves occurs at water rates in excess of 250 gallons per minute, this being due to the drop in discharge pressure when the speed is maintained constant. Up to full load, therefore, there is very little choice between the two methods of governing from considerations of economy; but from the general performance of the machine, it appears desirable to install a pump governor.

From curves C it will also be noted that there is very little difference in the economy when pumping cold water at 80 degrees F. or warm water at 170 degrees F.

A decrease in the exhaust pressure results in the increased economy shown on the upper right-hand figure of curves D for about full-load water rate.

By fitting a pressure gage to the steam ring it is possible to determine the amount of steam used per hour from the gage readings and the number of nozzles open, with the aid of the upper left-hand figure on curves D.

A high-pressure centrifugal pump should not be run for any extended period with the discharge closed, for the internal friction will raise the temperature of the churned water to an unsafe value. In starting, the steam turbine should be warmed and brought up to full speed quickly, and the valve on the pump discharge opened as soon as the required pressure is reached. The fact that a turbine-driven centrifugal pump can be quickly started from a cold condition and put into service constitutes one of its advantages.

Before starting, the pump should be fully primed, and it is preferable so to place the pump that the water will flow to it under a slight head. This latter condition becomes a necessity when hot water is pumped.

With the high speeds at which these pumps revolve the steam and other pipes should be so arranged and secured that no part will have a natural vibration period equal to the rotative speed.

The performance of this pump may be compared with that of a simplex reciprocating boiler-feed pump recently tested at the Experiment Station* by assuming a piston speed of 75 feet per minute for the simplex pump to correspond to a water rate of 250 gallons per minute for the centrifugal pump.

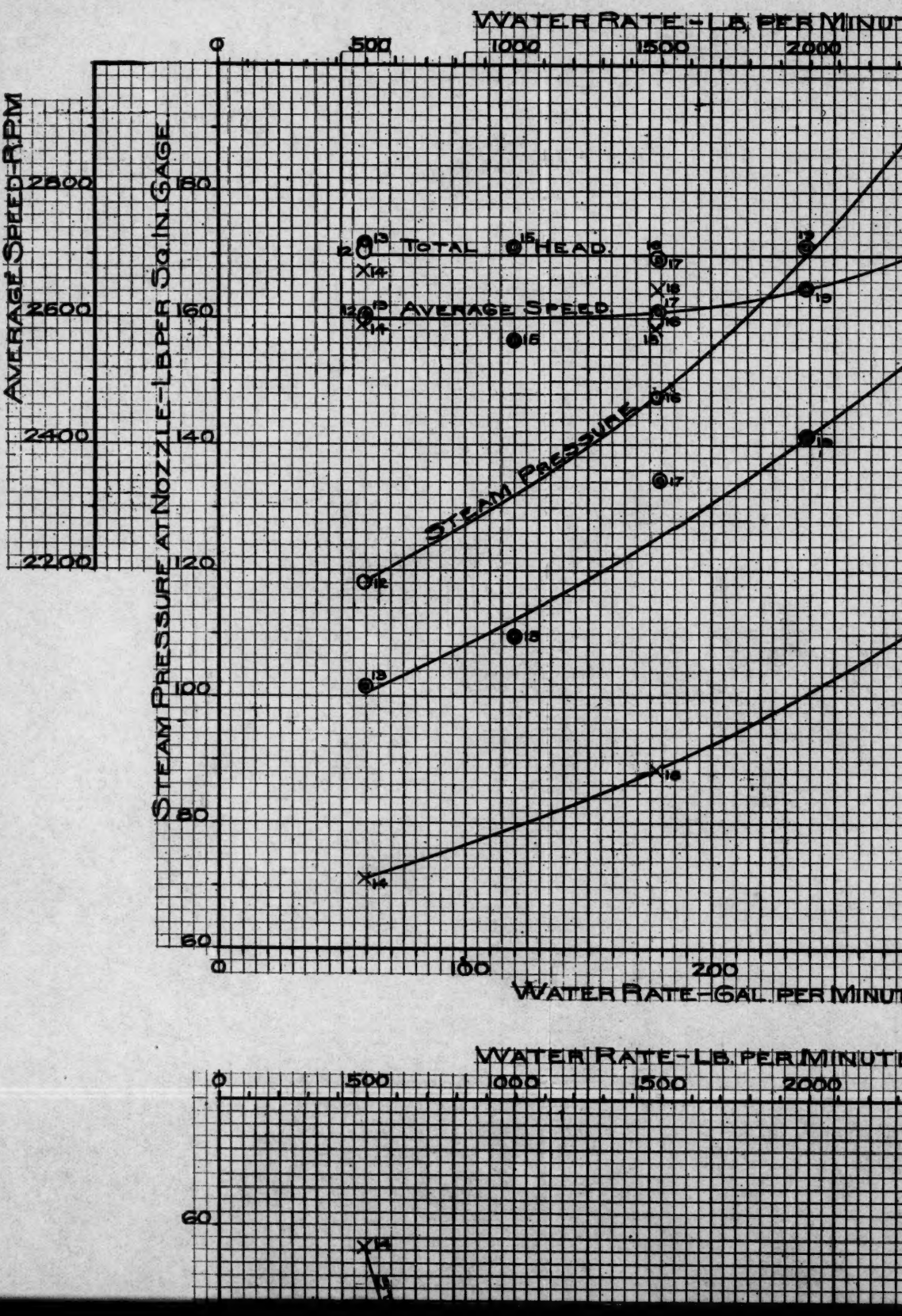
| Per cent. of full load. | Steam per 1,000 pounds water pumped, pounds. | | |
|---|---|---|---|
| | Simplex reciprocating. | | Centrifugal. |
| | Cold water. | Warm water. | Cold or warm water. |
| 33 | 34.2 | 58.5 | 55 |
| 67 | 29.6 | 37.0 | 32 |
| 100 | 26.2 | 29.4 | 27 |
| 133 | 29.0 | 31.2 | 26 |

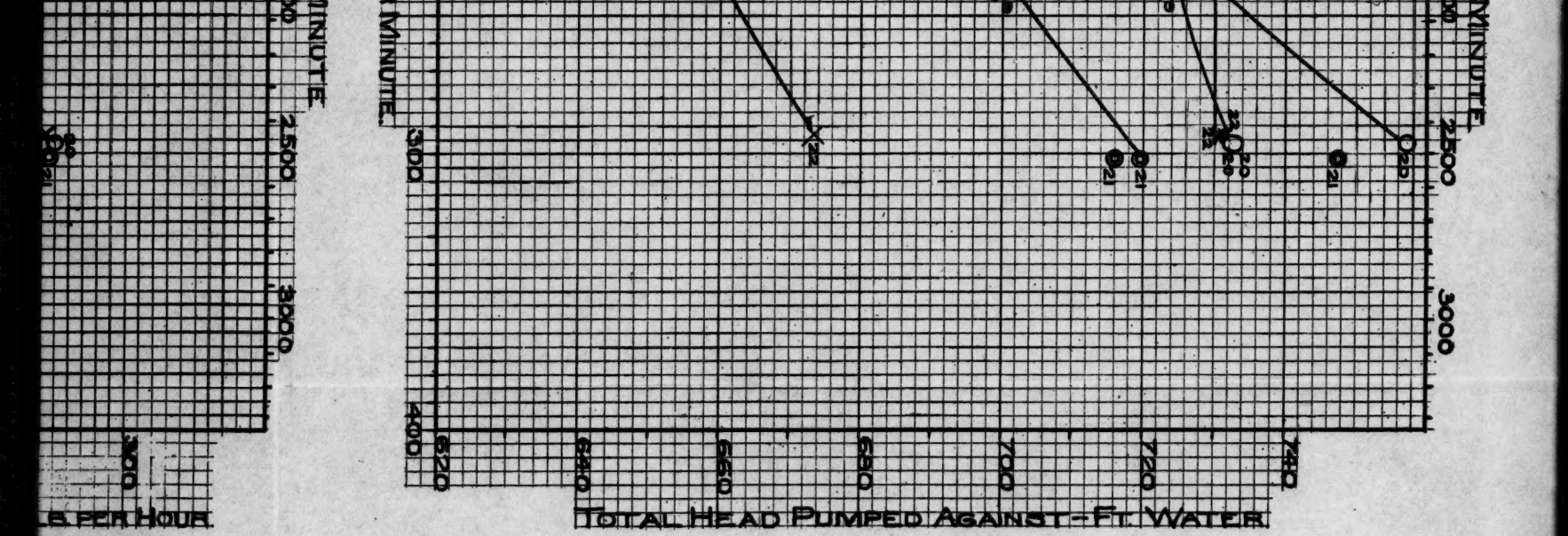
From the foregoing comparison it will be seen that when pumping warm water there is little difference between the economy of this centrifugal pump and the simplex pump. For cold water, up to 33 per cent. of full load, the simplex pump is more efficient, but at 60 per cent. to 100 per cent. the efficiency is about the same. From 100 per cent. to 133 per cent. the centrifugal pump is more efficient. As the values in the last column are based upon a careful regulation of the number of nozzles in use according to the load, the comparison will be less favorable to the centrifugal pump at light loads if

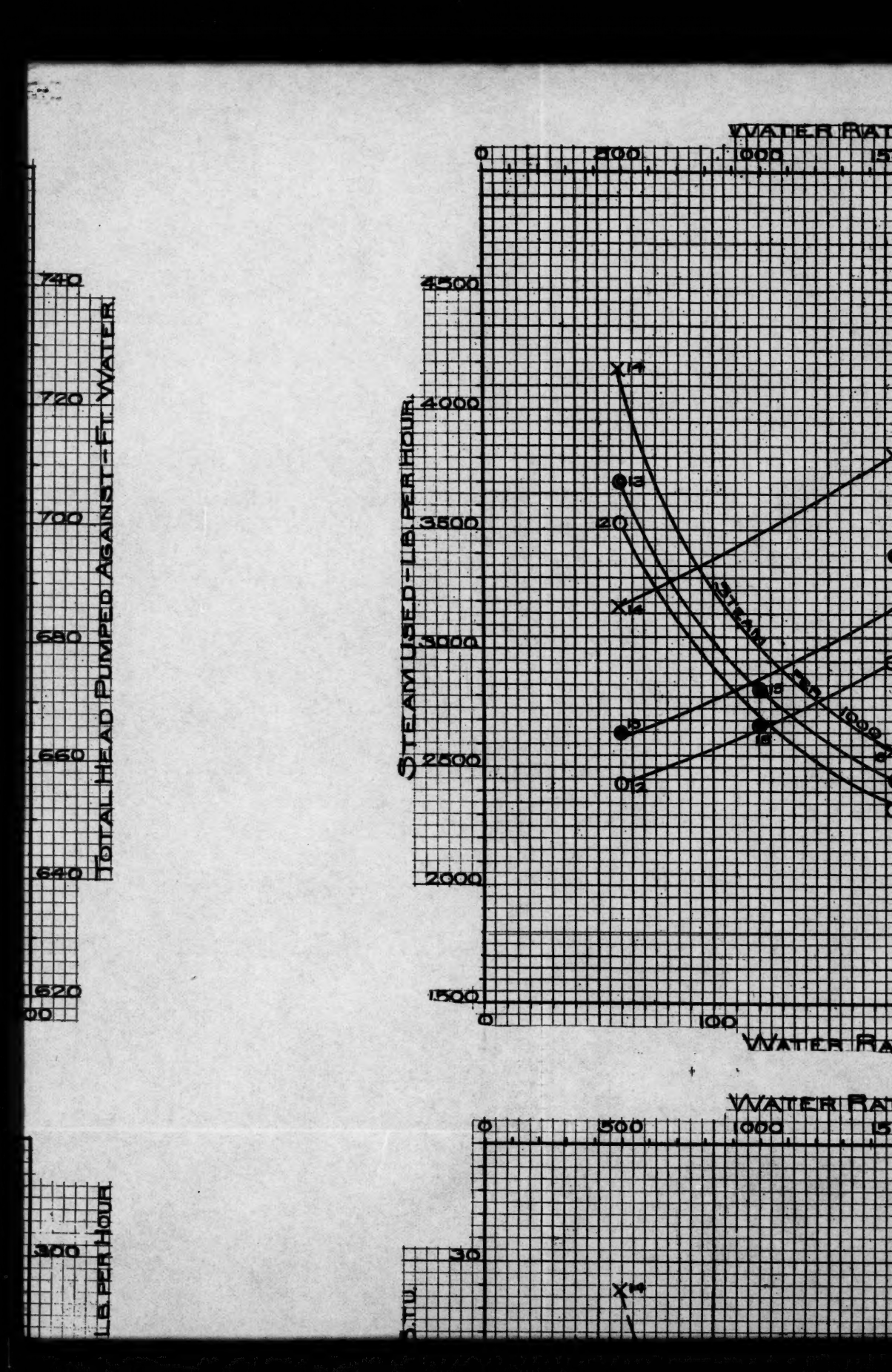
* JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXIV, No. 3, p. 882, August, 1912.

such regulation is not carried out. On the other hand, the test of the simplex pump was conducted under the most favorable circumstances as to the condition of the working parts of the simplex pump. Reciprocating pumps decrease steadily in efficiency due to wear, while the centrifugal pump will probably change very little in this respect. Further, the centrifugal pump being of an experimental type, certain slight mechanical changes will doubtless improve its efficiency.

On the whole, therefore, the foregoing comparison, while interesting, can not be considered as a definite comparison of results which may be expected to obtain under service conditions.


WATER RATE - LB. PER MINUT
0
500
1000
1500
2000
AVERAGE SPEED-RPM
2800
2600
2400
2200
STEAM PRESSURE AT NOZZLE-LB PER SQ. IN. GAGE
160
140
120
100
80
60
TOTAL HEAD.
AVERAGE SPEED.
STEAM PRESSURE
0
100
200
WATER RATE - GAL. PER MINU
WATER RATE - LB. PER MINUT
0
500
1000
1500
2000
60


2500
3000
300
400
620
640
660
680
700
720
740
TOTAL HEAD PUMPED AGAINST - FT. WATER
STEAM USED - LBS. PER HOUR.


740
720
700
680
660
640
620
TOTAL HEAD PUMPED AGAINST - FT. WATER
300
LB. PER HOUR
WATER RAT
0
500
1000
4500
4000
3500
3000
2500
2000
1500
STEAM USED - LB. PER HOUR
X14
13
20
12
15
100
WATER RA
WATER RAT
0
500
1000
30
X14
B.T.U.

R RATE - LBS PER MINUTE.

1500 2000 2500 3000

STEAM PER 1000 POUNDS WATER PUMPED - LBS.

120 100 80 60 40 20 0

X22 X18 X17 X16

O21 O20 O18 O17 O16

1000 x POUNDS OF WATER

200 300 400

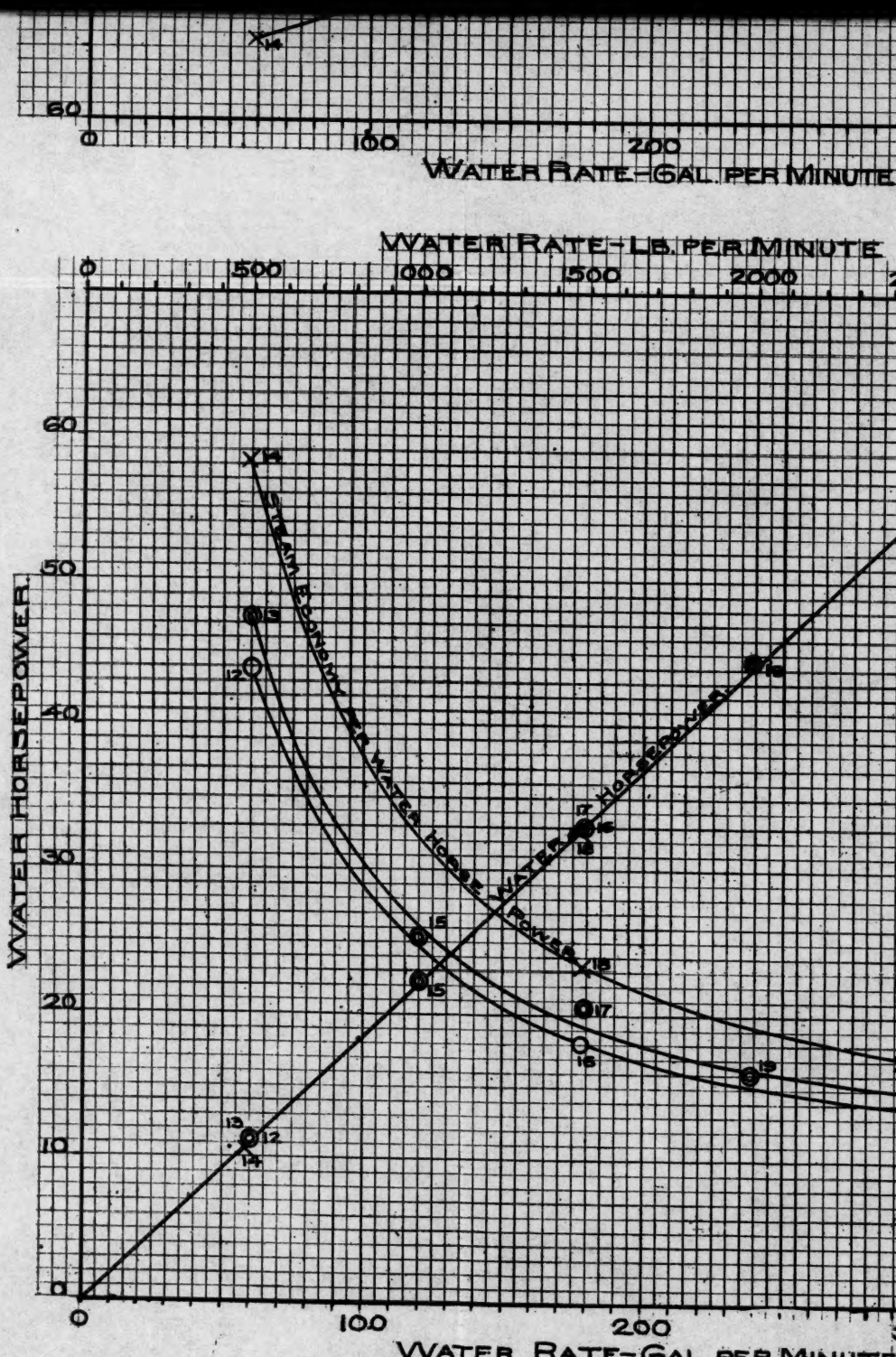
ER RATE - GAL. PER MINUTE.

R RATE - LBS PER MINUTE.

1500 2000 2500 3000

6000

ER MINUTE.




NOTE:-

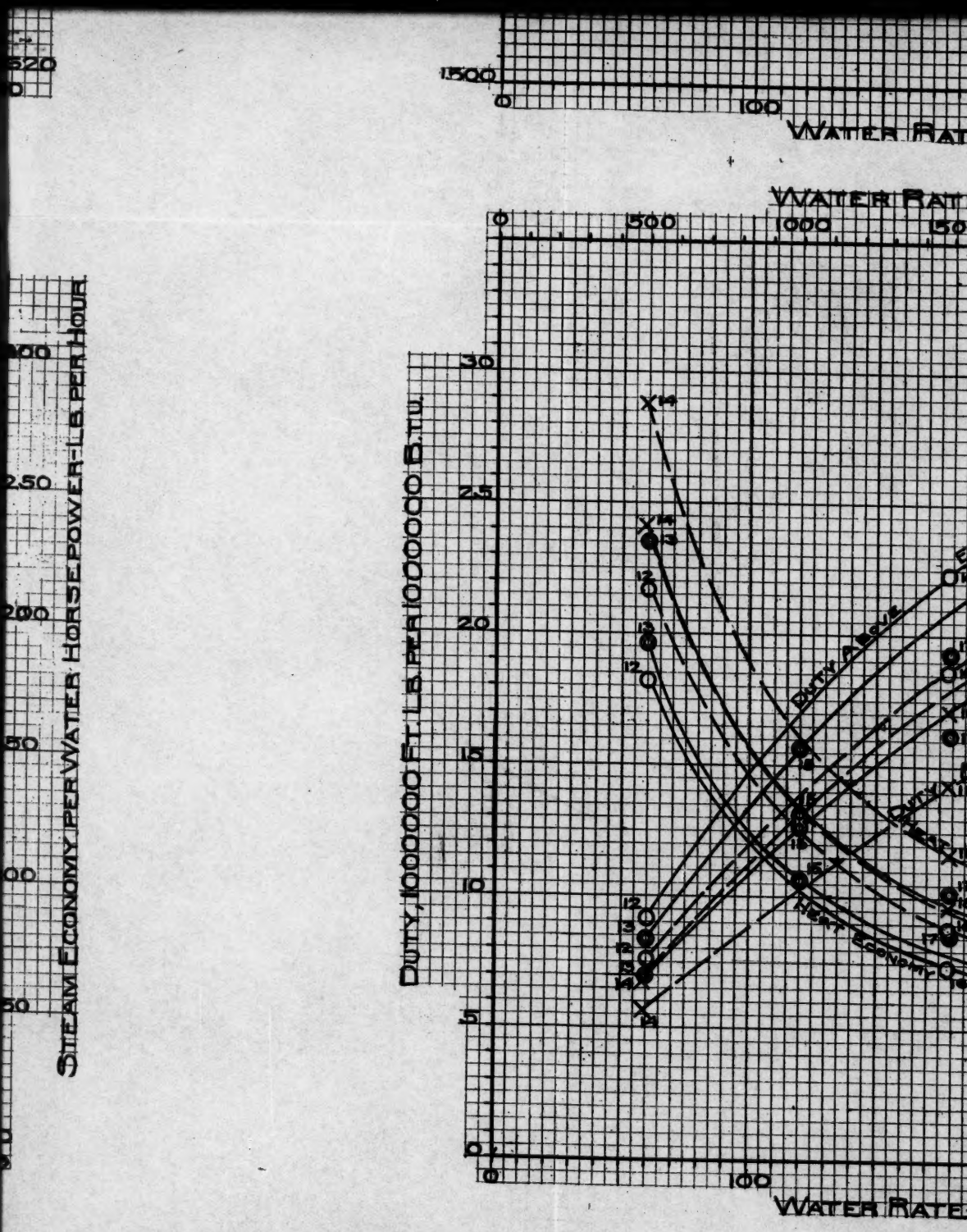
POINTS MARKED THUS O CORRESPOND TO FOUR NOZZLES OF

" " " ◎ " " FIVE "

" " " X " " EIGHT "

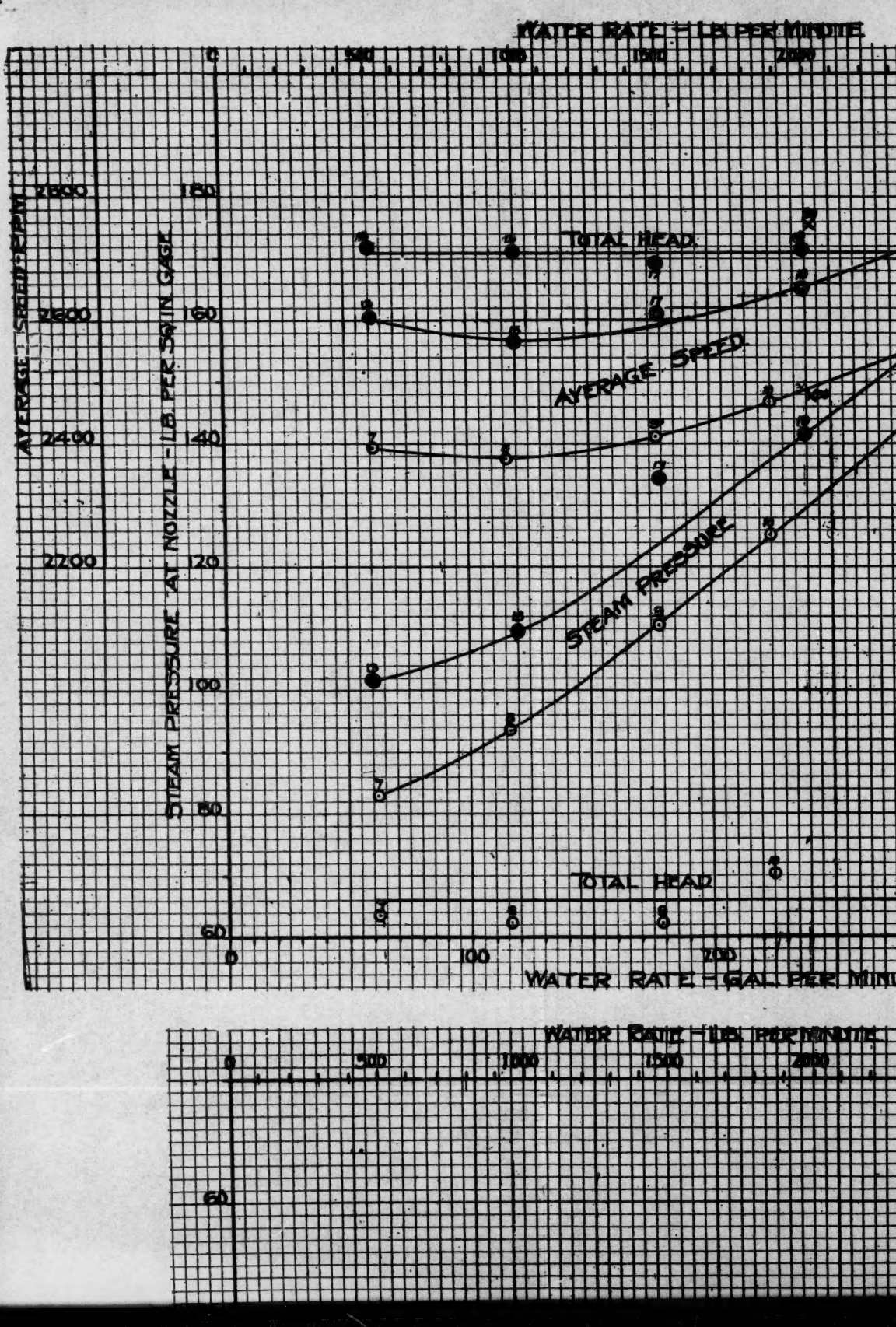
SMALL NUMBERS REFER TO RUN NUMBER.

Curves A.—Effect of Varyi



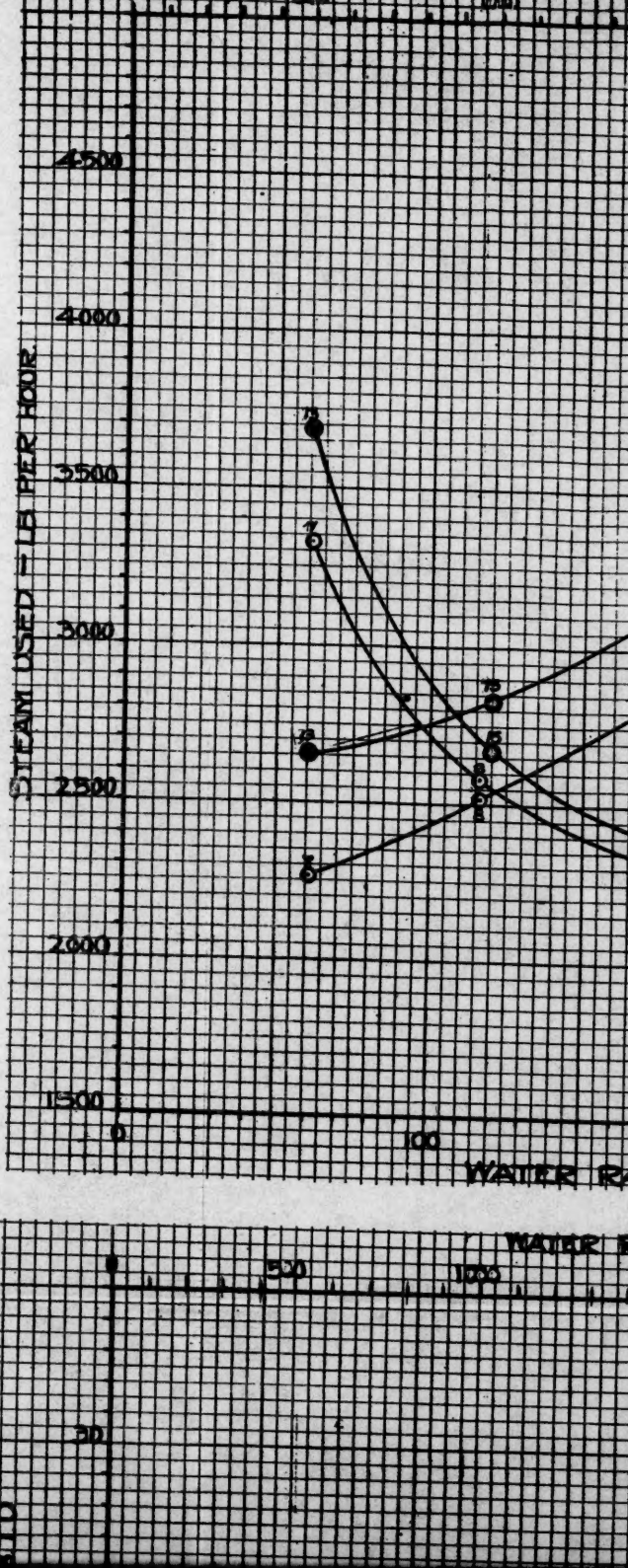

A.—Effect of Varying the Number of Noz

Nozzles Open.


WATER RATE - LB. PER MINUTE
0
500
1000
1500
2000
AVERAGE SPEED - R.P.M.
2800
2600
2400
2200
STEAM PRESSURE AT NOZZLE - LB. PER SQ. IN. GAGE
180
160
140
120
100
80
60
TOTAL HEAD
AVERAGE SPEED
STEAM PRESSURE
TOTAL HEAD
0
100
200
WATER RATE - GAL. PER MIN.
WATER RATE - LB. PER MINUTE
0
500
1000
1500
2000
60

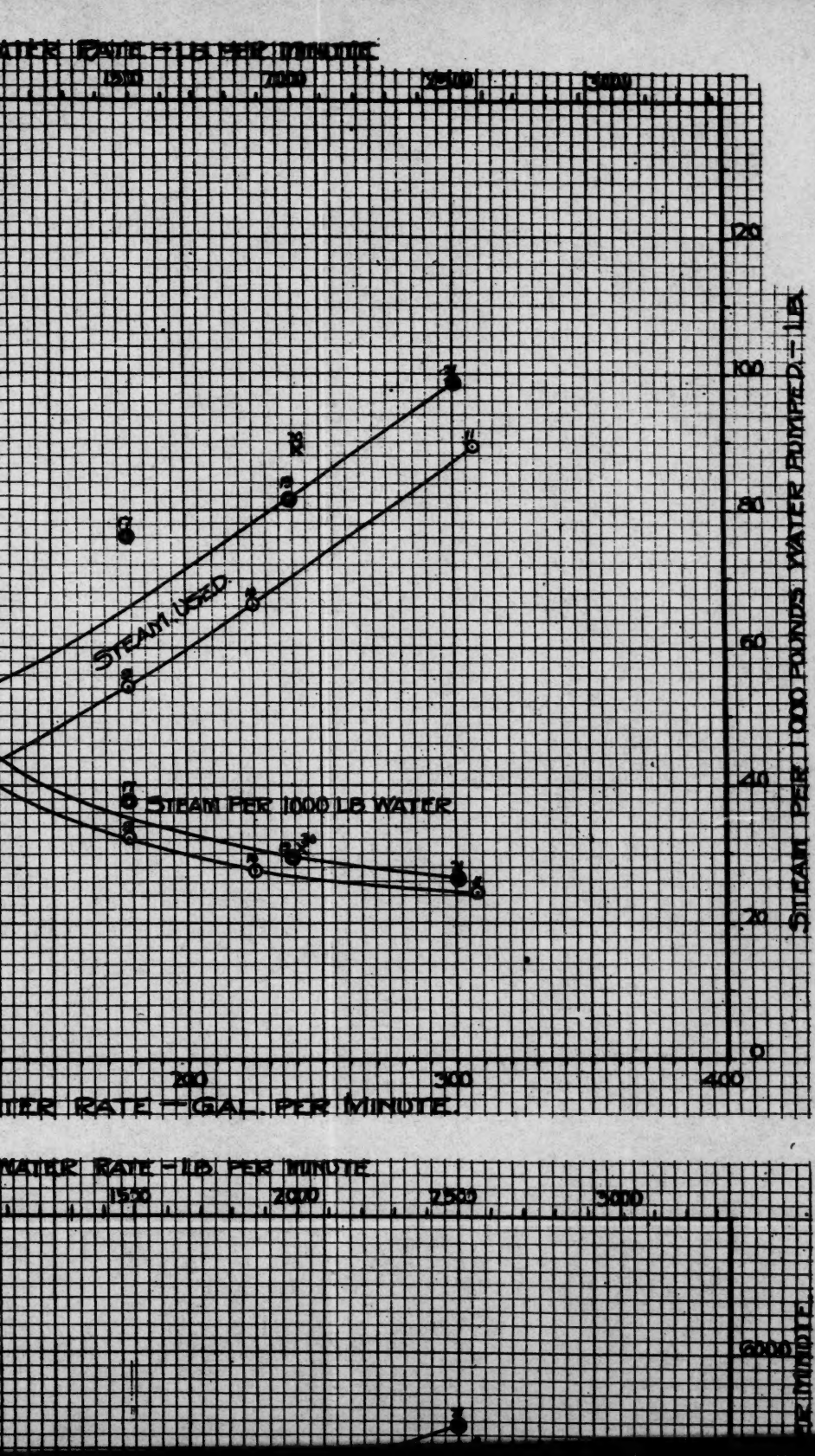
MINUTE
2500
3000
740
720
700
680
660
640
620
TOTAL HEAD PUMPED AGAINST — FT. WATER
300
400
R MINUTE
MINUTE
2500
3000
300
STEAM USED — LB PER HOUR

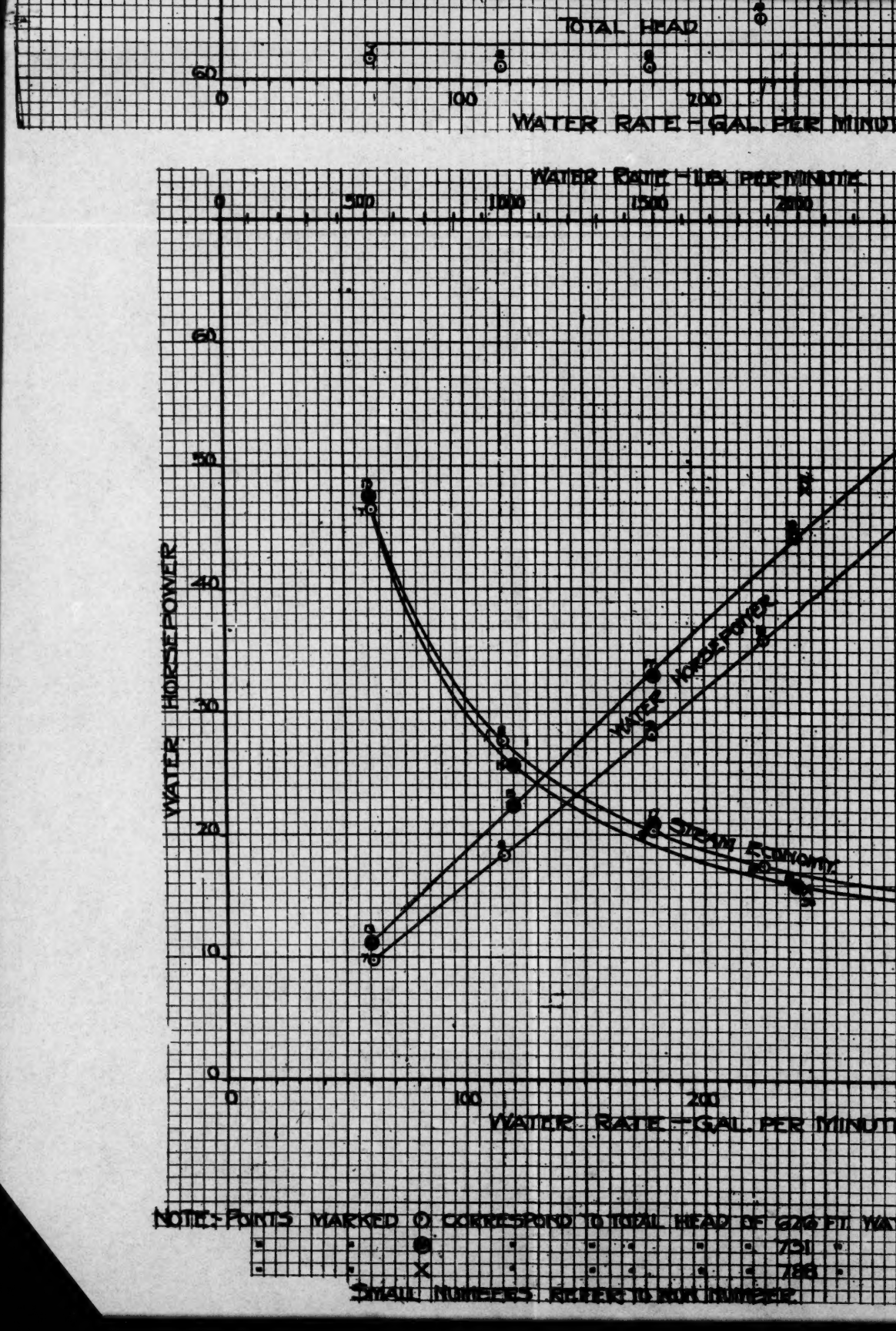
740
720
700
680
660
640
620
400
TOTAL HEAD PUMPED AGAINST — FT. WATER.

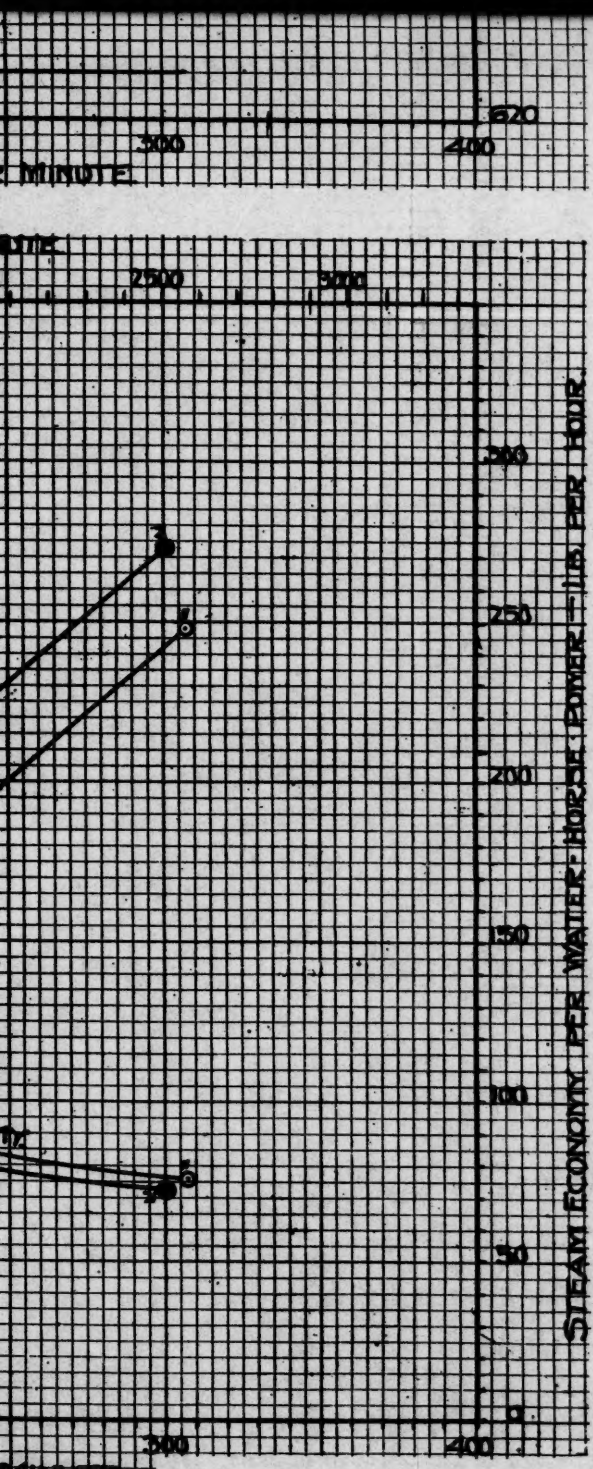

WATER
300
4500
4000
3500
3000
2500
2000
1500
0
100
STEAM USED — LB PER HOUR.
WATER R

300
LBS PER HOUR.

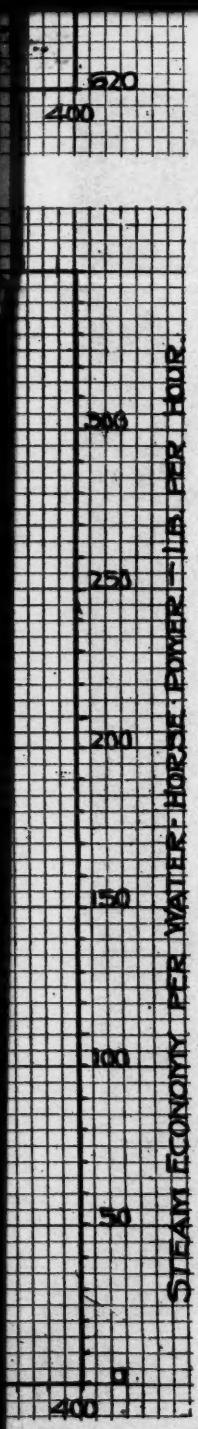
WATER
500
1000
30

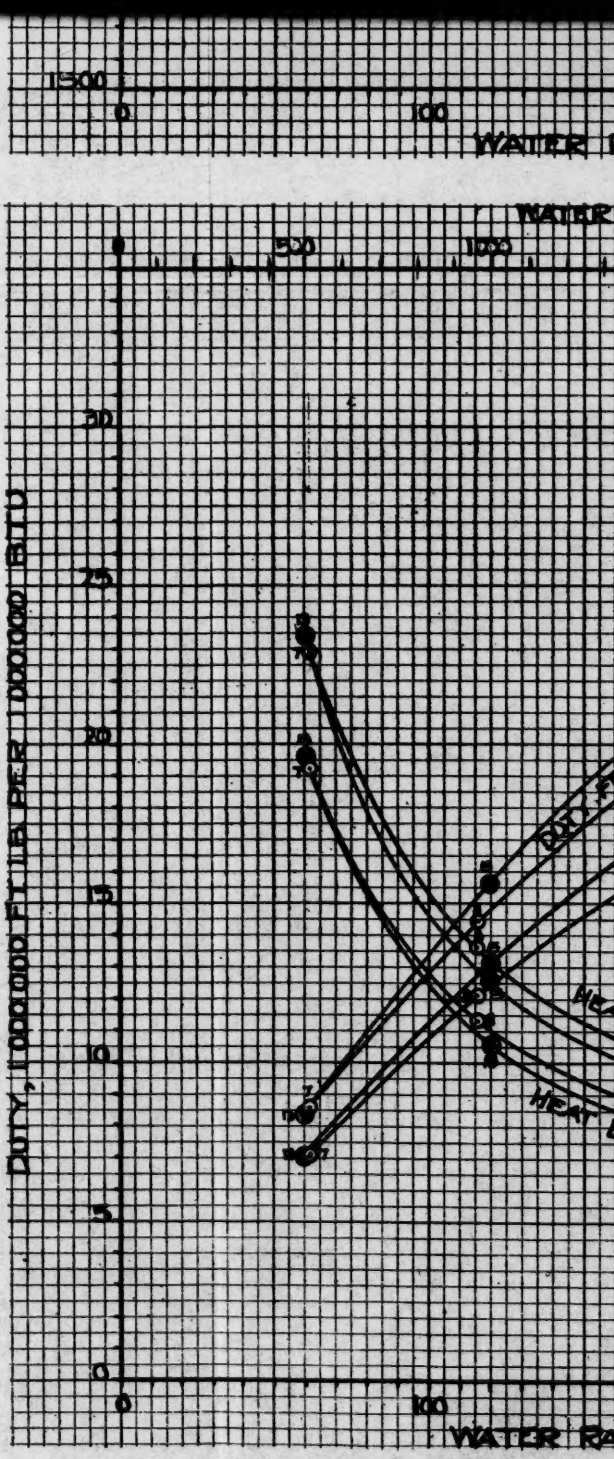

ATER RATE = LB PER MINUTE
1500
2000
2500
3000
120
100
80
60
40
20
0
STEAM PER 1000 POUNDS WATER PUMPED = LB
STEAM USED
STEAM PER 1000 LB WATER
200
300
400
ER RATE = GAL. PER MINUTE
WATER RATE = LB PER MINUTE
1500
2000
2500
3000
6000


TOTAL HEAD
60
0
100
200
WATER RATE - GAL. PER MINUTE
WATER RATE - GAL. PER MINUTE
0
500
1000
1500
2000
60
50
40
30
20
10
0
WATER HORSE POWER
WATER HORSE POWER
STEAM ECONOMY
0
100
200
WATER RATE - GAL. PER MINUTE
NOTE:- POINTS MARKED O CORRESPOND TO TOTAL HEAD OF 626 FT. WA
731
788
SMALL NUMBERS REFER TO RUN NUMBER.

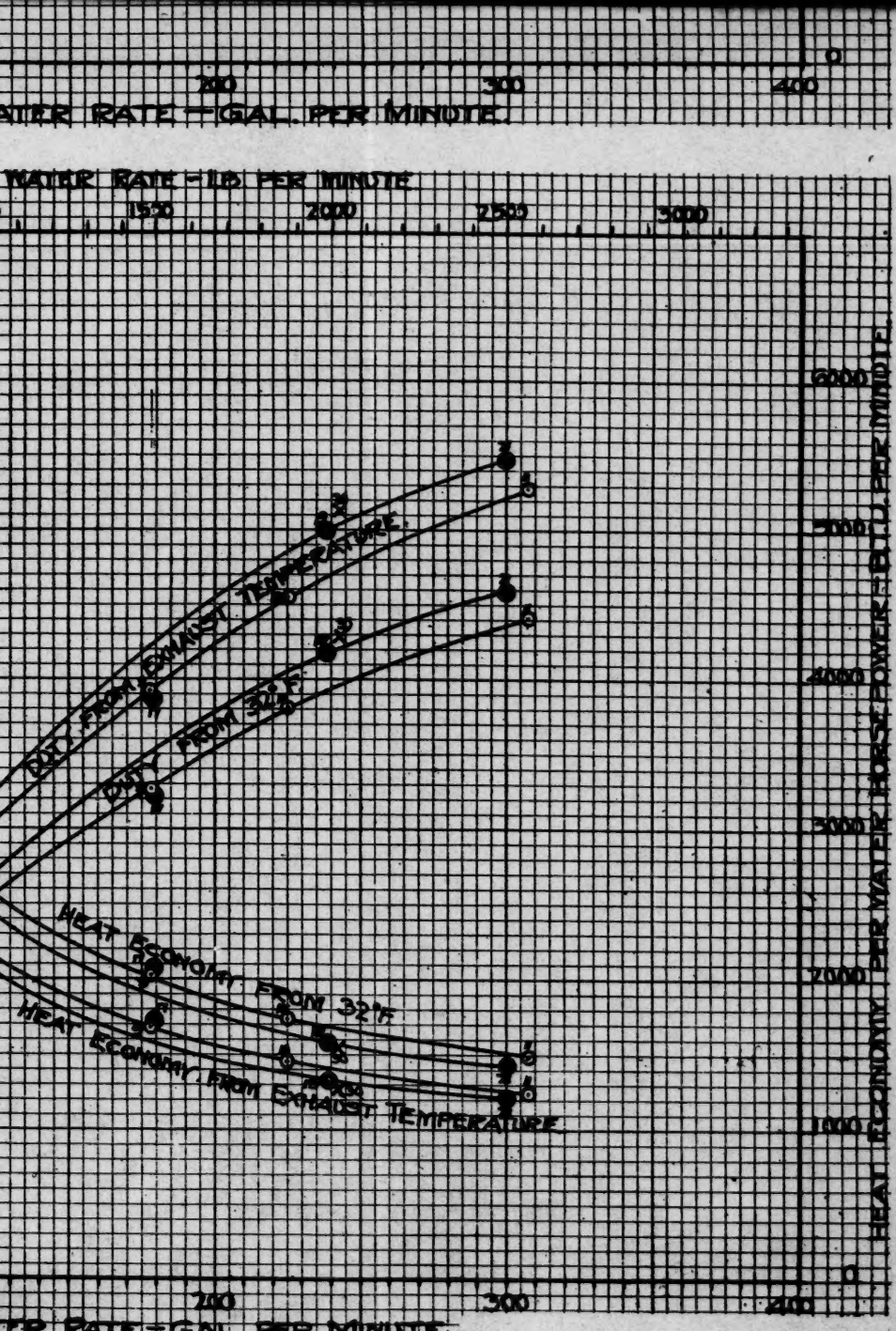



Curves B.—Effec

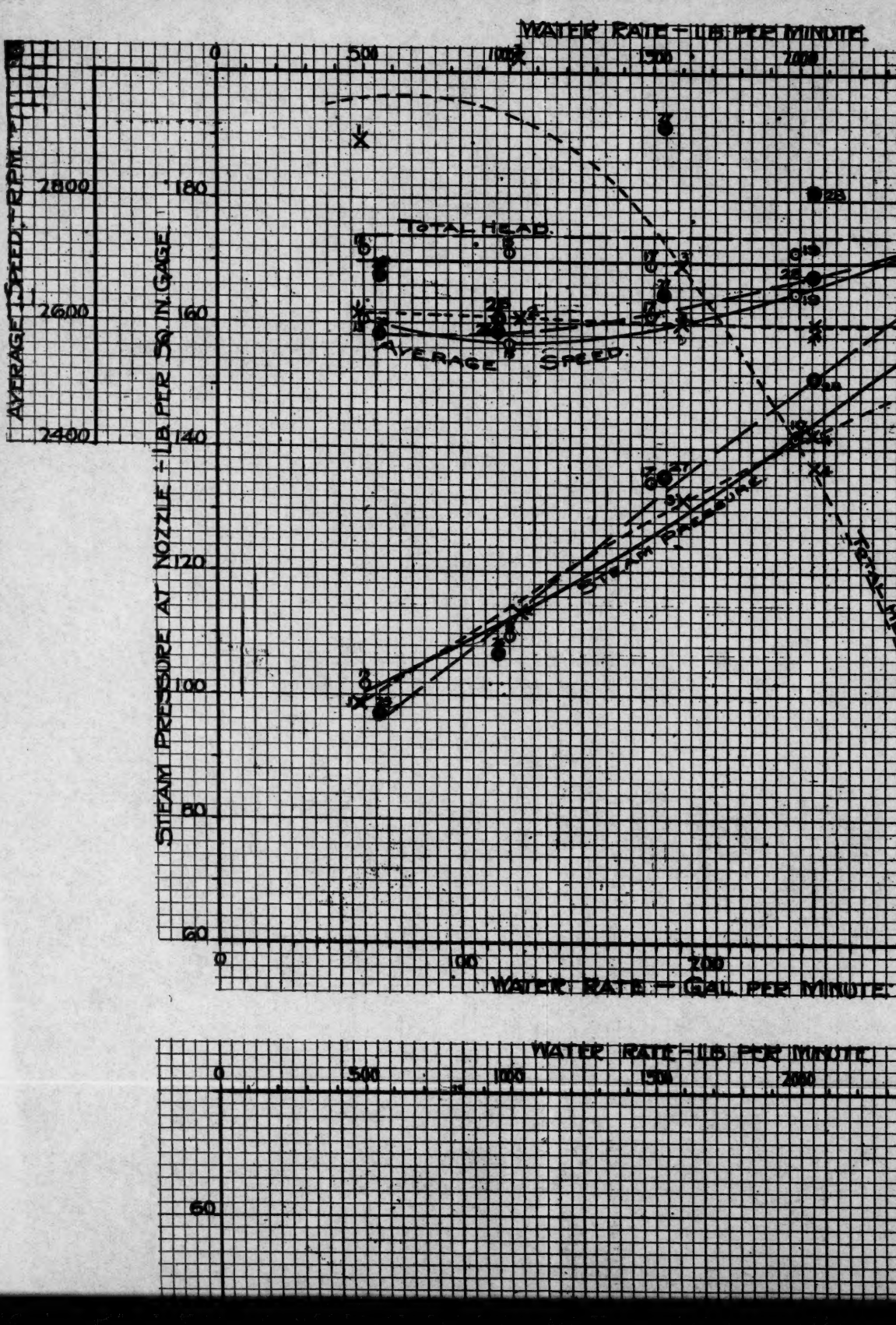



Curves B.—Effect of Varying the Hea

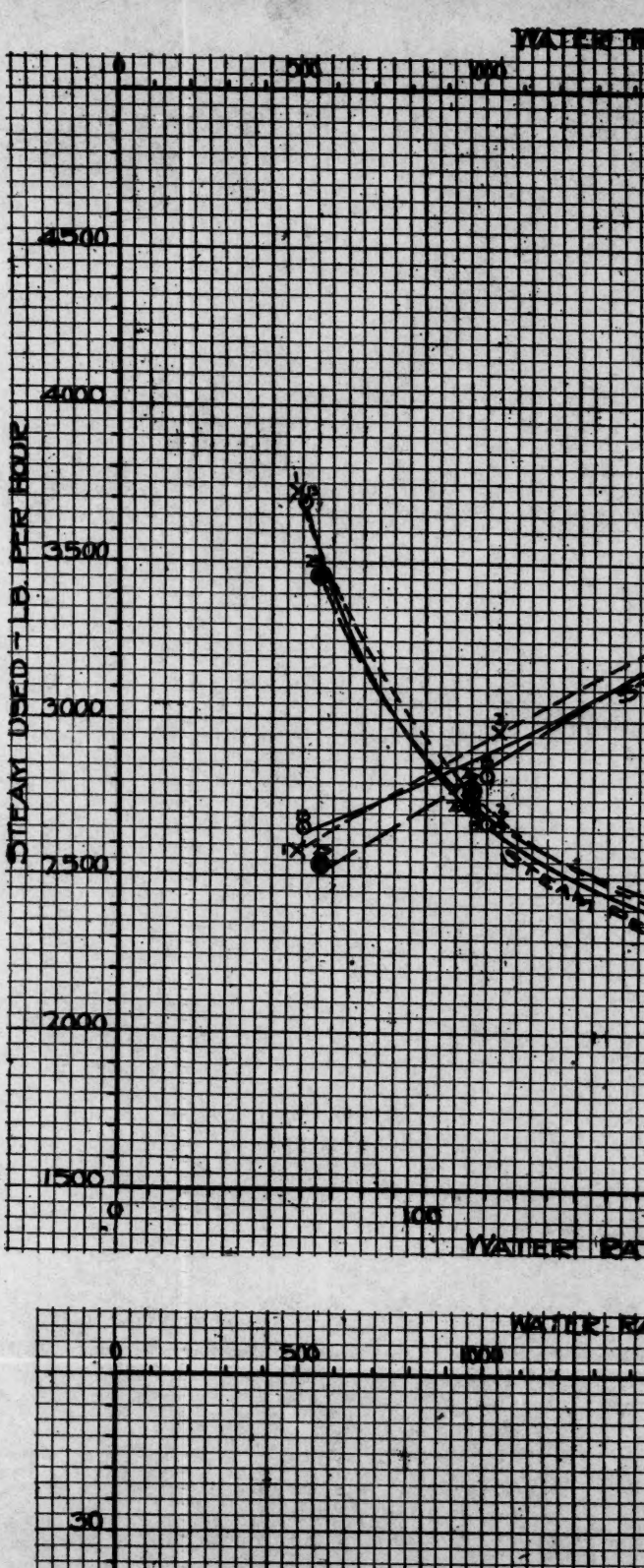

WATER RATE - LB. PER MINUTE.
1500
2000
2500
3000
HEAT ECONOMY PER WATER HORSE POWER - B.T.U. PER MINUTE.
6000
5000
4000
3000
2000
1000
0
HEAT ECONOMY FROM EXHAUST TEMPERATURE.
HEAT ECONOMY FROM 32°F.
200
300
400
WATER RATE - GAL. PER MINUTE.

Head.

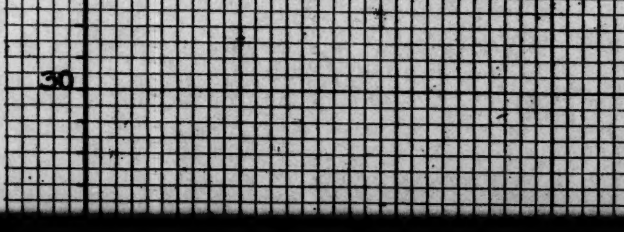

WATER RATE - LB PER MINUTE
0
500
1000
1500
2000
AVERAGE SPEED - RPM
2800
2600
2400
180
160
140
120
100
80
60
STEAM PRESSURE AT NOZZLE - LB PER SQ IN GAGE
TOTAL HEAD
AVERAGE SPEED
STEAM PRESSURE
0
100
200
WATER RATE - GAL PER MINUTE
WATER RATE - LB PER MINUTE
0
500
1000
1500
2000
60

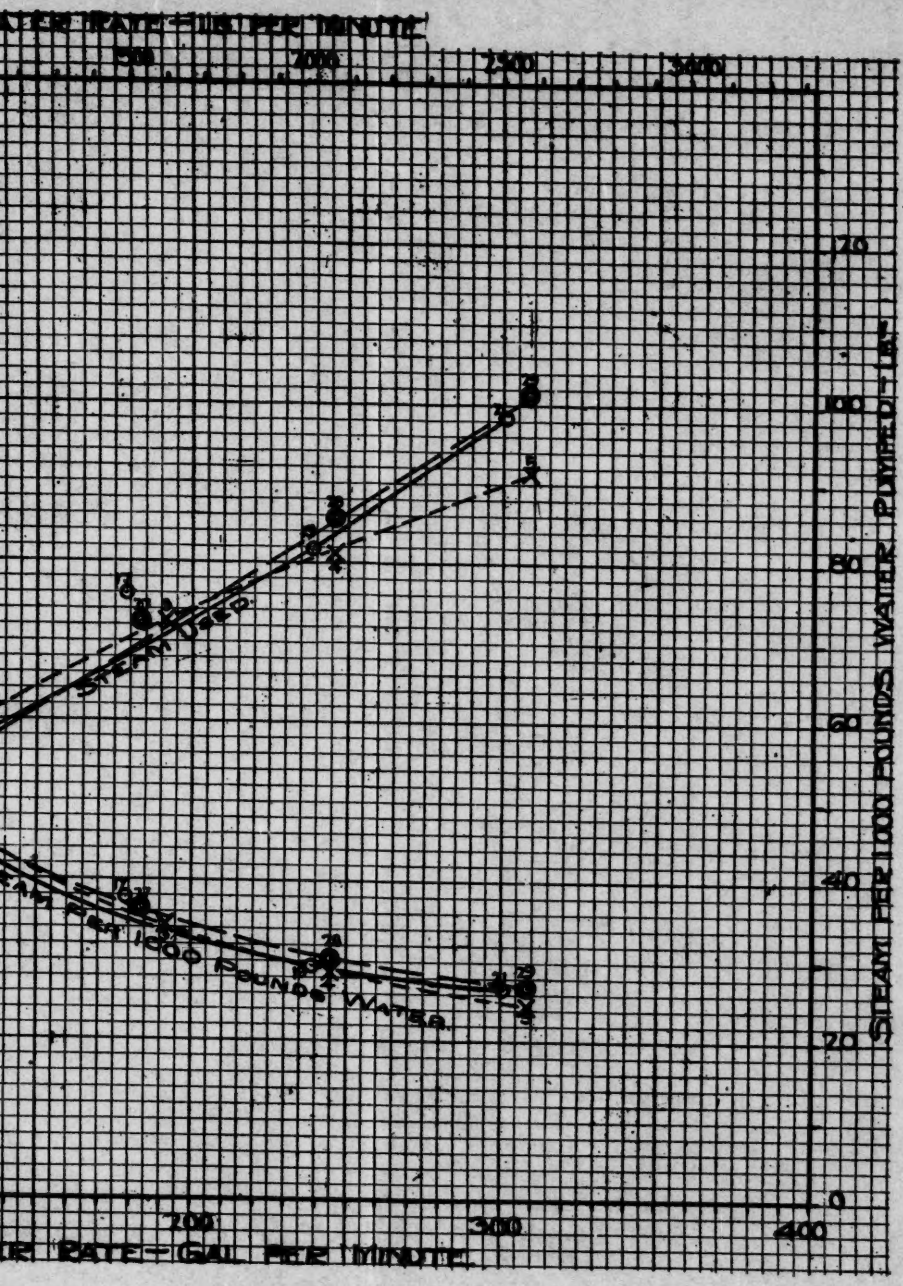
NOTE:
2500
3000
740
720
700
680
660
640
620
300
400
TOTAL HEAD PUMPED AGAINST — FT. WATER.
MINUTE:
NOTE:
2500
3000
300
PER HOUR.

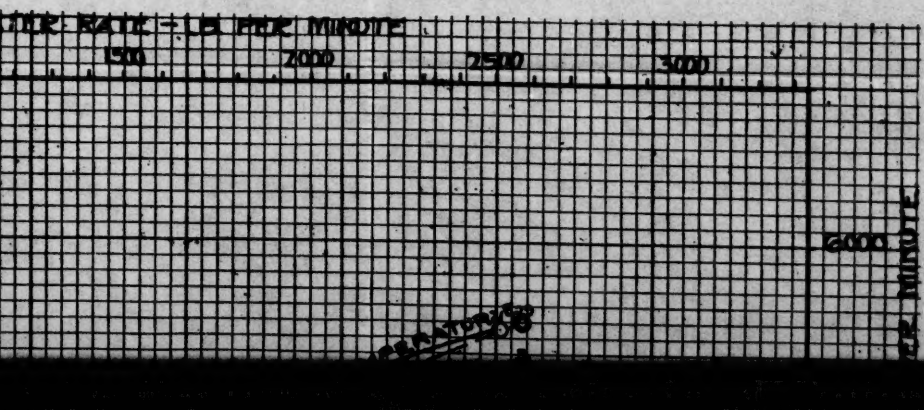
720
700
680
660
640
620
TOTAL HEAD PUMPED AGAINST — FT. WATER.

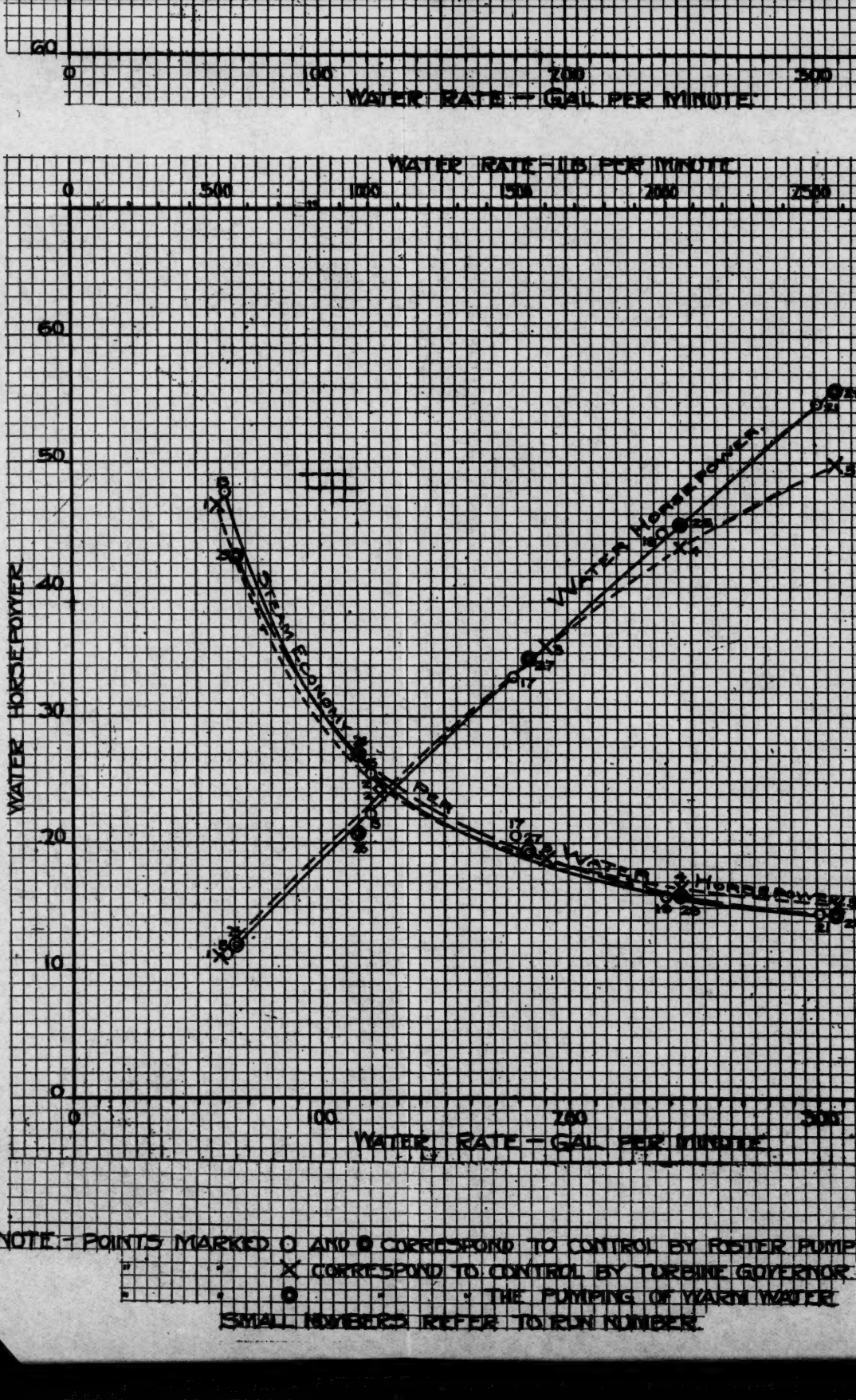

WATER R
0
500
1000
4500
4000
3500
3000
2500
2000
1500
STEAM USED — LB. PER HOUR.
WATER RA

300
LB. PER HOUR.

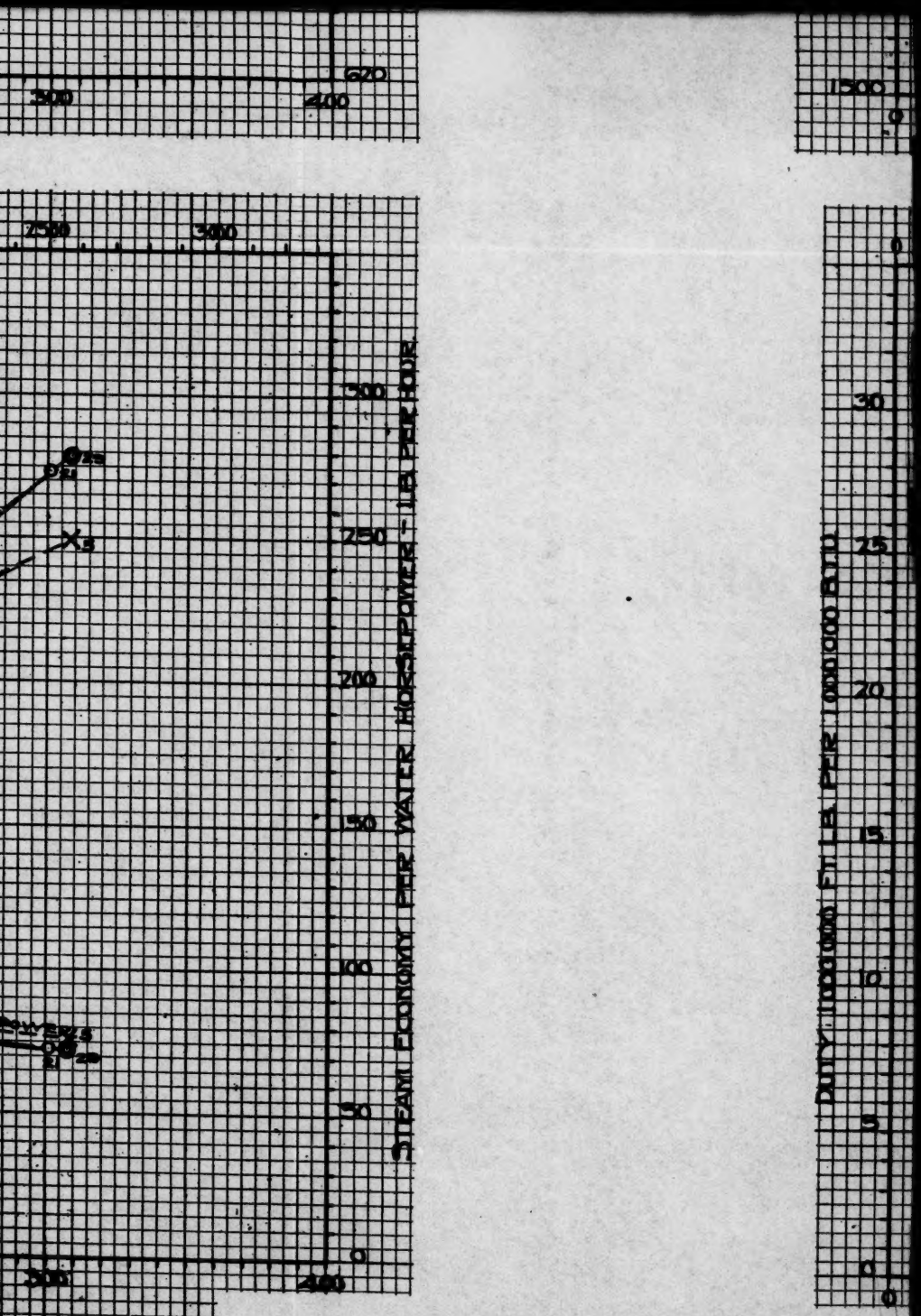

WATER RA
0
500
1000
30


ER RATE = LBS PER MINUTE
1500
2000
2500
3000
STEAM PER 1000 POUNDS WATER POWER = LBS
160
140
120
100
80
60
40
20
0
STEAM PER 1000 POUNDS WATER
700
3000
400
ER RATE = GAL PER MINUTE

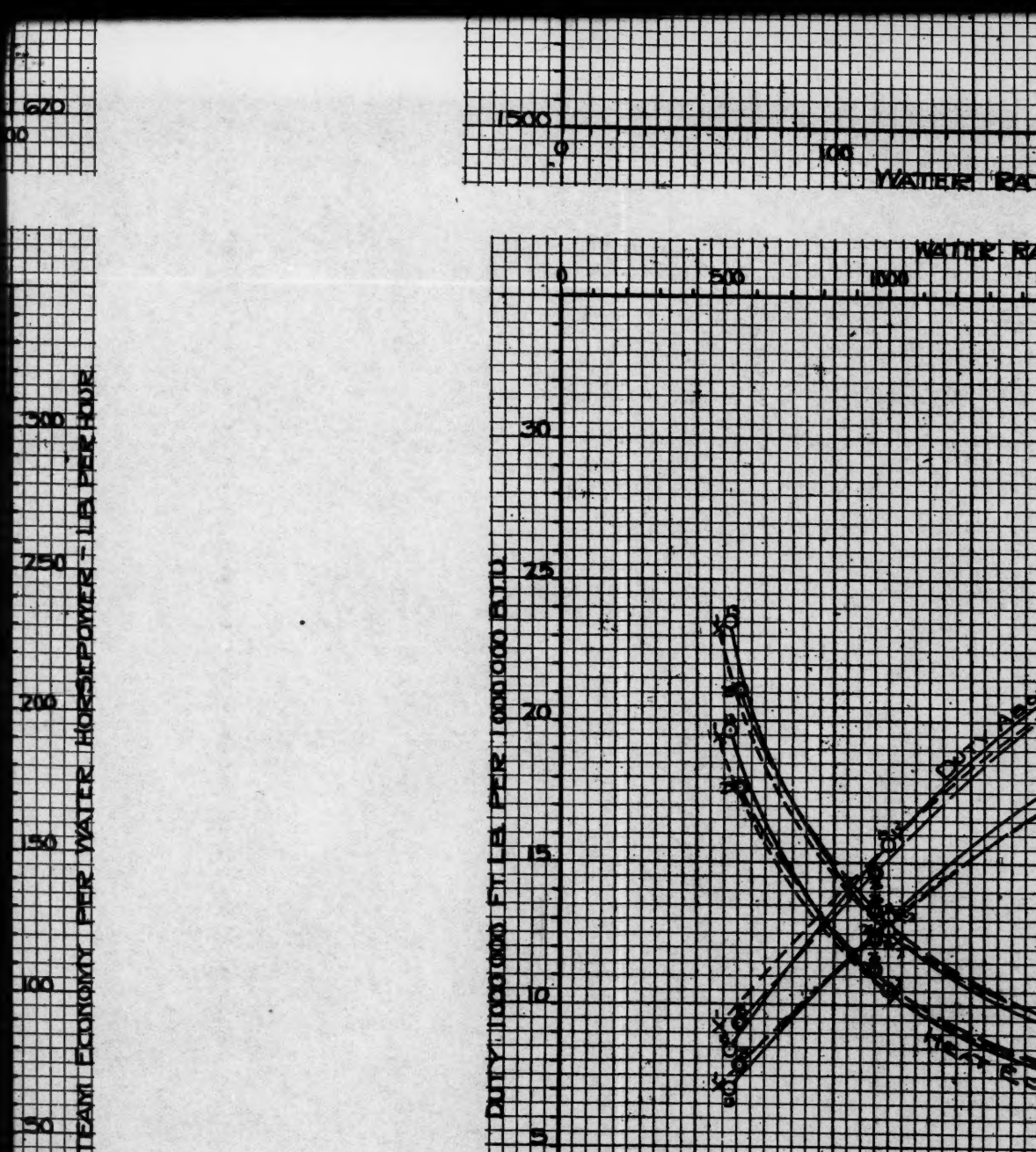

ER RATE = LB PER MINUTE
1500
2000
2500
3000
6000
ER MINUTE



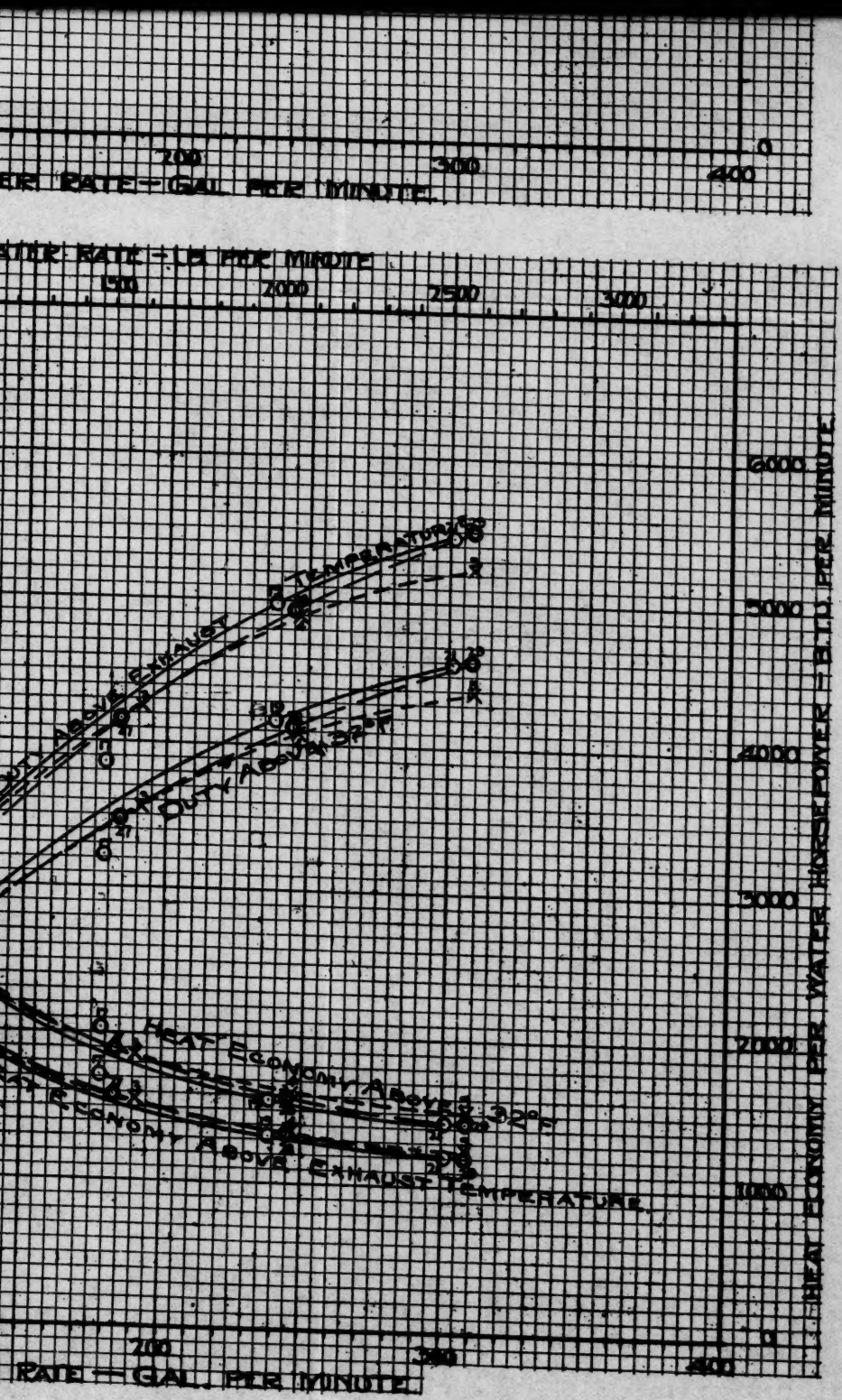

NOTE - POINTS MARKED O AND ● CORRESPOND TO CONTROL BY FOSTER PUMP
" " X CORRESPOND TO CONTROL BY TURBINE GOVERNOR
" " ● " " THE PUMPING OF WARM WATER
SMALL NUMBERS REFER TO RUN NUMBER.



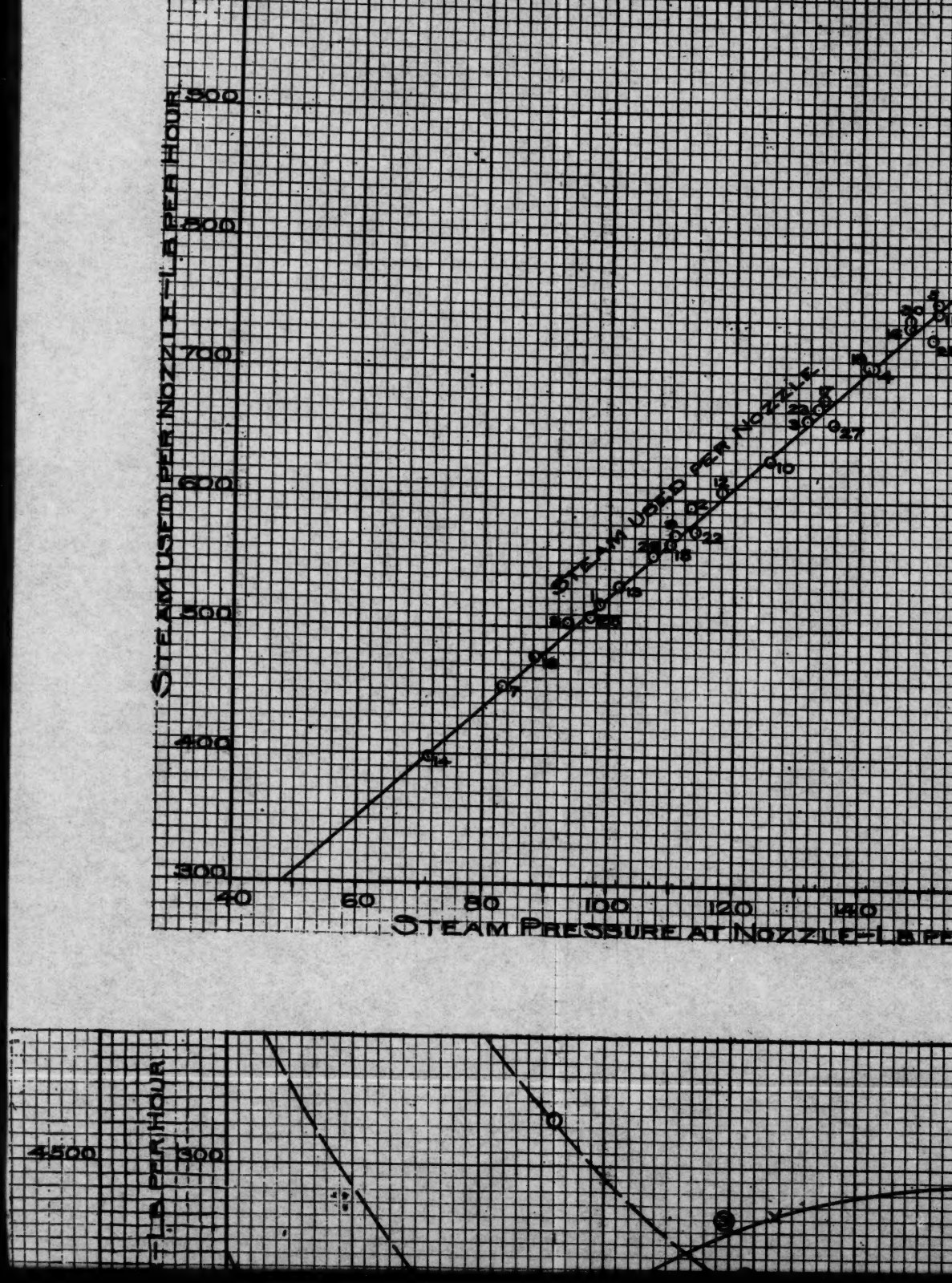

Curves C.—Effect of Meth

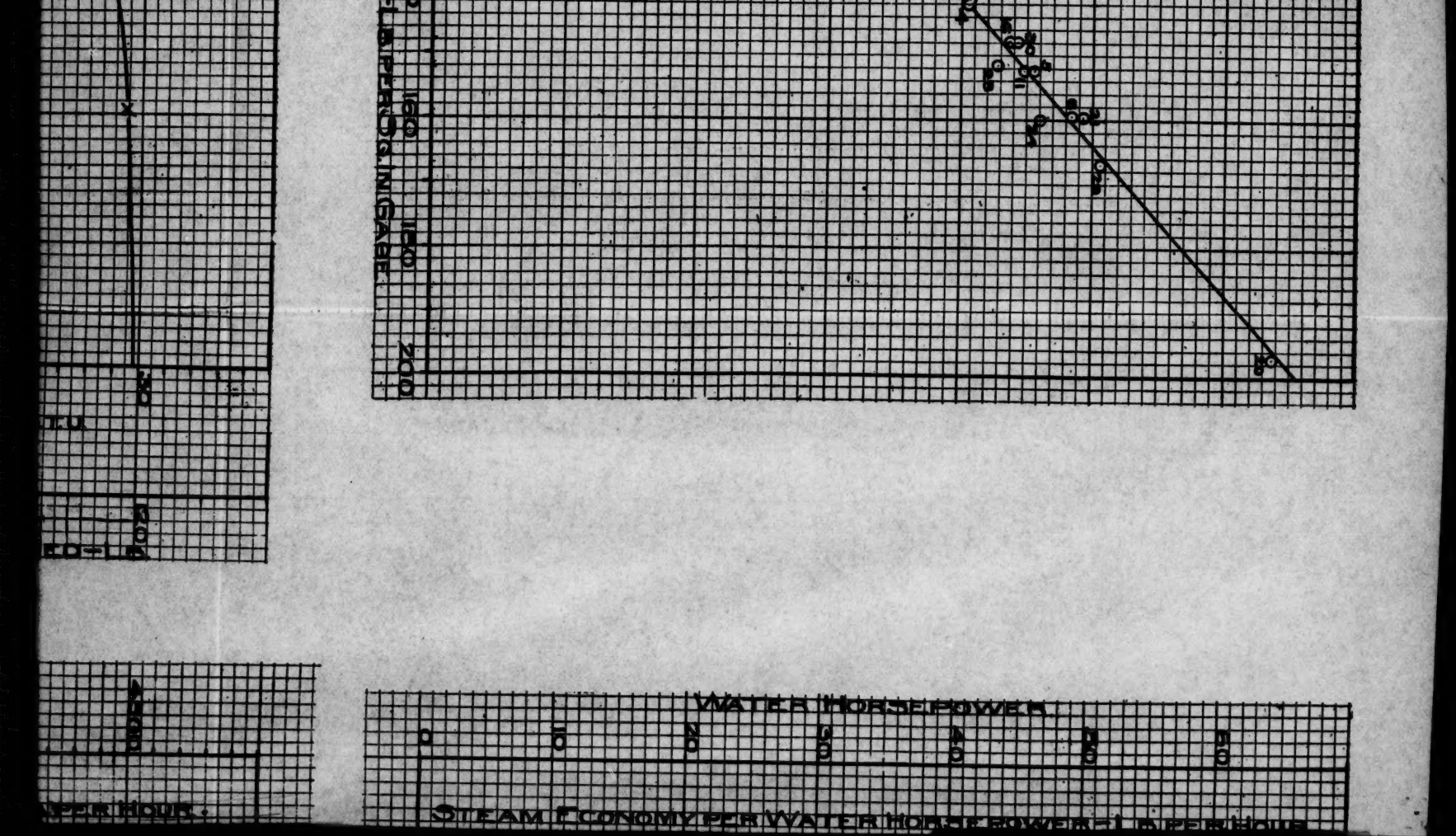


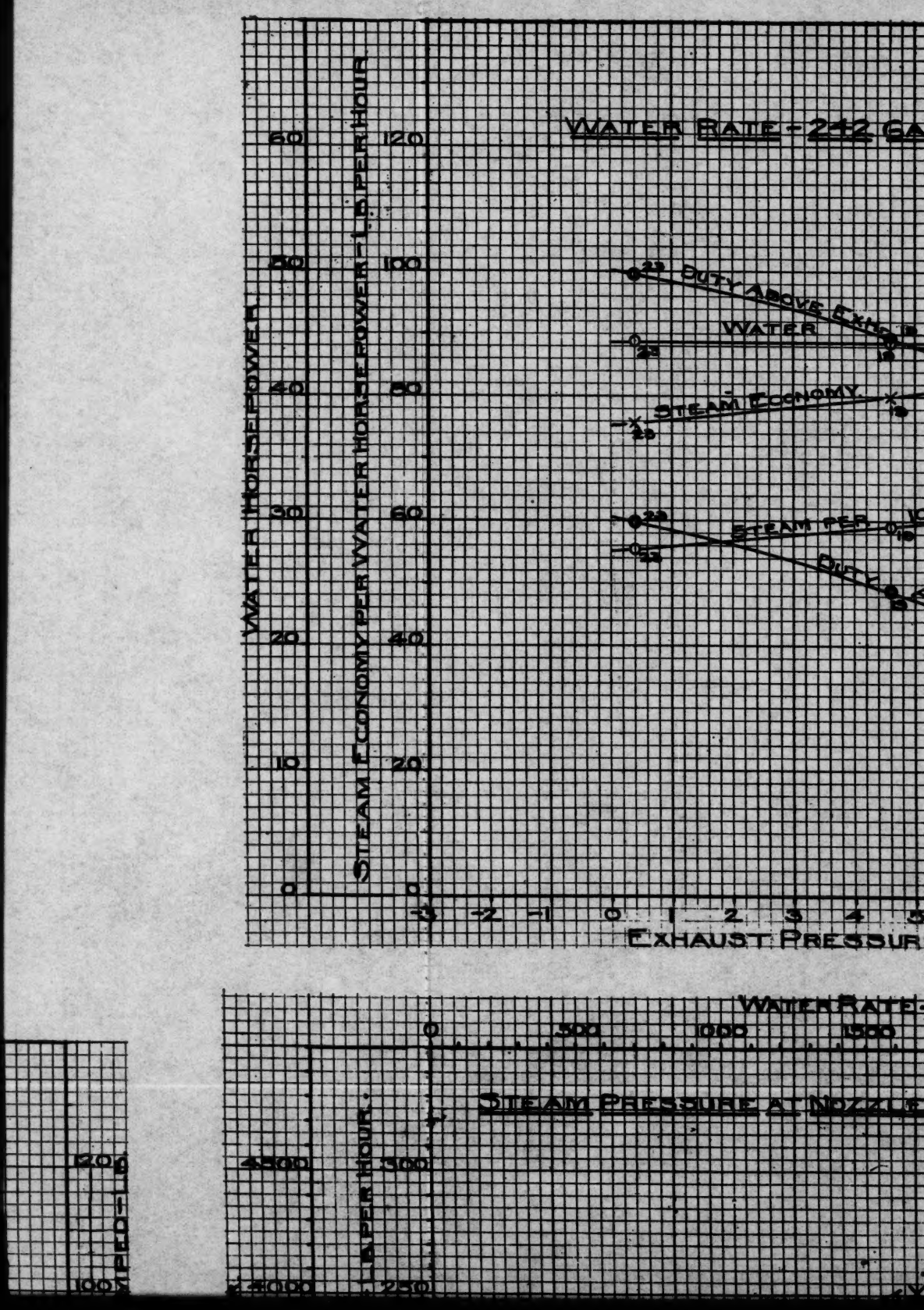

ves C.—Effect of Method of Governing a




g and of Suction Temperature.


STEAM USED PER NOZZLE-LB PER HOUR
500
600
700
300
400
40
60
80
100
120
STEAM PRESSURE AT NOZZLE-LB
LB PER HOUR
4500
300


WATER HORSE POWER
STEAM ECONOMY PER WATER HORSE POWER = LB PER HOUR


WATER RATE - 2+2 GA
DUTY ABOVE EXH
WATER
STEAM ECONOMY
STEAM PER
DUTY
WATER HORSE POWER
STEAM ECONOMY PER WATER HORSE POWER - LBS PER HOUR
60
40
30
20
10
0
120
100
80
60
40
20
0
-3
-2
-1
0
1
2
3
4
EXHAUST PRESSUR
WATER RATE
0
500
1000
1500
STEAM PRESSURE AT NOZZLE
4500
4000
300
250
LB PER HOUR
120
100
SPEED - L

...2 GALLONS PER MINUTE.

...IN HORSEPOWER.

...TEMPERATURE.

...1000 LBS. WATER.

...ABOVE 32° F.

...SURE - LB. PER SQ. IN. GAGE.

...RATE - LB. PER MINUTE.

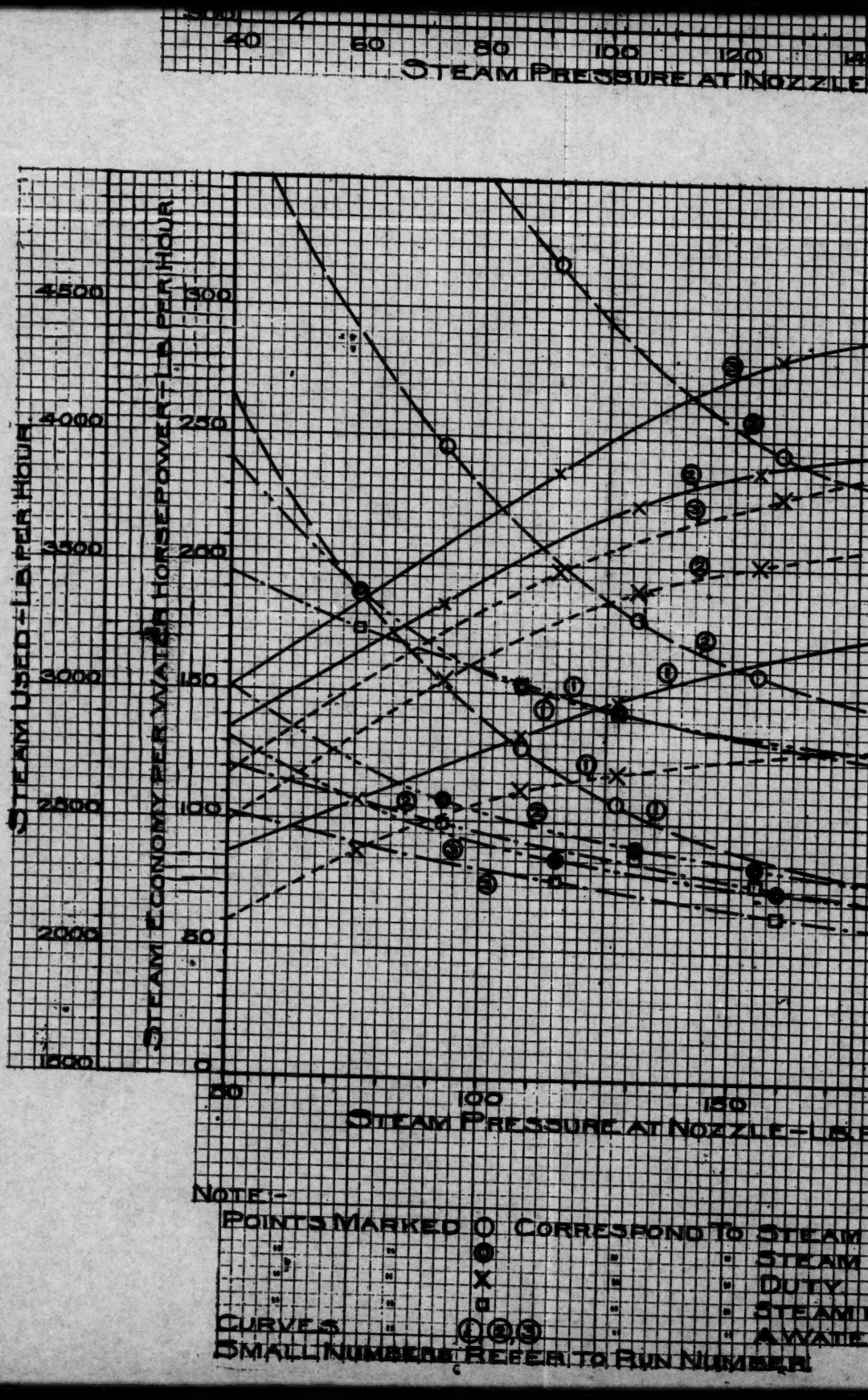
...LUE - 2.25 LB. PER SQ. IN. GAGE.

...TEMPERATURE.

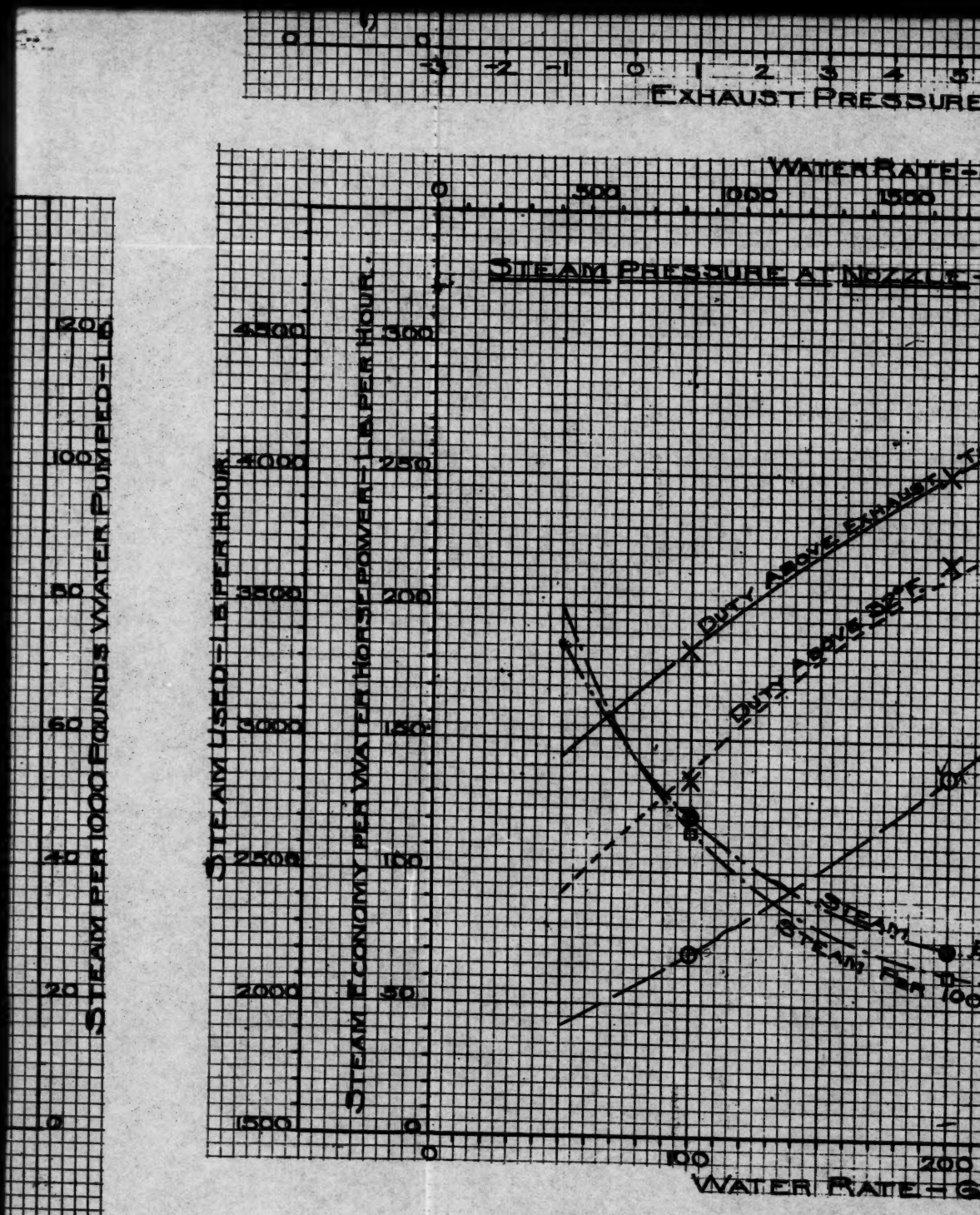
OUTPUT 1000000 B.T.U. PER HOUR

STEAM PER 1000 POUNDS WATER PUMPED - LBS.

IMPROVEMENT - %

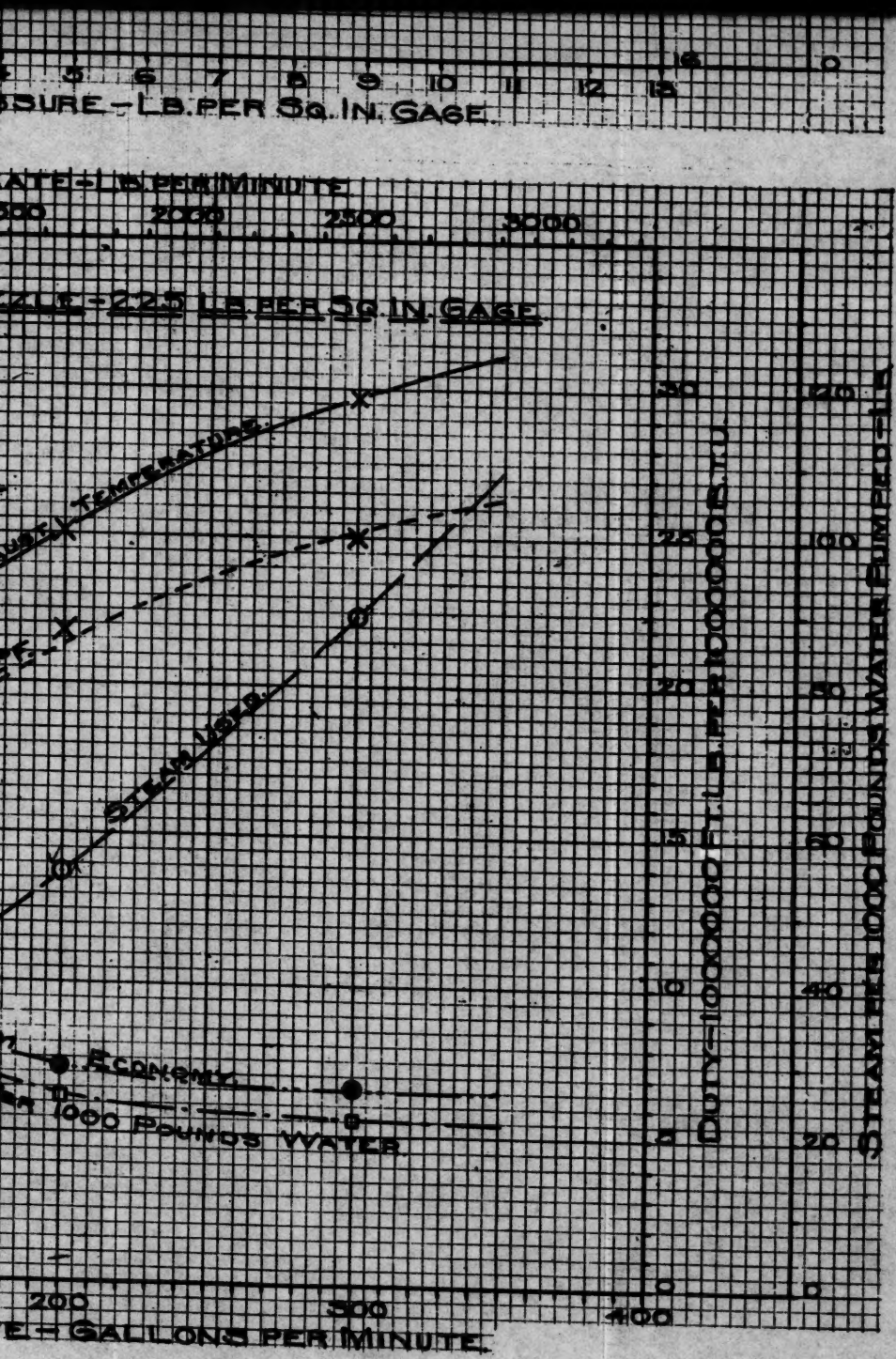

40
60
80
100
120
STEAM PRESSURE AT NOZZLE
STEAM USED-LB PER HOUR
4500
4000
3500
3000
2500
2000
1500
STEAM ECONOMY PER WATER HORSEPOWER-LB PER HOUR
300
250
200
150
100
50
0
50
100
150
STEAM PRESSURE AT NOZZLE-LB
NOTE:-
POINTS MARKED O CORRESPOND TO STEAM
STEAM
DUTY
STEAM
A WATE
CURVES
SMALL NUMBERS REFER TO RUN NUMBERS

140
160
180
200
ZZLE-LBS PER SQ. IN GAGE.
30
25
20
15
10
5
0
DUTY-1000000 FT. LBS PER 100000 B.T.U.
120
100
80
60
40
20
0
STEAM USED-LBS PER HOUR
4500
4000
3500
3000
2500
2000
1500
200
250
-LBS PER SQ. IN. GAGE.
TEAM USED.
EAM ECONOMY PER WATER HORSEPOWER.
UTY.
EAM PER 1000 POUNDS WATER PUMPED.
WATER RATE OF 100 200 AND 300 GALLONS PER MINUTE
RI




Curves D.—Effec
and Incre

ILONS PER MINUTE RESPECTIVELY.




Effect of Varying Exhaust Pressure
ncreasing Nozzle Pressure.

# THE FACTOR OF ASSURANCE.

## A MEANS OF MAKING THE BEST OF YOUR WAY AT SEA.

By Commander Victor Blue, U. S. Navy.

In making a long voyage at sea without an available coal pile near the route the commander of a vessel is naturally much concerned about her coal endurance. Especially is this true if the length of the route approaches closely the radius of action of the vessel. The proper thing to do, of course, is to steam at the most economical speed. But what is the most economical speed? It may be defined as the speed which will permit of the vessel going the greatest distance with the coal she has on board. There are so many varying elements and conditions that affect the problem that it is quite impracticable to determine the most economical speed for once and all and to make use of it under all circumstances. The captain of a vessel about to start on a long voyage will select the speed which he believes to be the most economical as determined from data calculated by the engine designers or obtained from experience on previous voyages. Similarly, he will obtain the tons of coal required per mile, and then the total for the run for comparison with the amount in the bunkers. The most important question then is, What should be considered a safe margin? The allowance made for this margin should not be too large on account of the limitations it would place on the operations of the ship, nor, for obvious reasons, should it be too small. The determination of the proper margin depends upon many variable elements and conditions, the principal ones being:

(*a*) Kind and quality of coal.

(*b*) Efficiency of boilers and engines.

(*c*) Condition of the ship's bottom.

(*d*) Displacement and trim.

(*e*) Condition of wind and sea that may be encountered.

Experience has shown that all these elements and conditions are variable quantities. Coal from the same mine will vary in efficiency according to the lapse of time since it was mined, and according to the weather exposure it has undergone. The efficiency of the boilers and machinery varies with the wear and tear of service, and with the repairs made from time to time.

The condition of the ship's bottom varies with the length of time out of dock, with the time spent at anchor in different waters, and with the distance steamed.

Displacement and trim vary from day to day, according to amount of coal used and bunkers from which taken.

The most potent factor is the weather to be expected. Seldom, if ever, does a ship making regular transoceanic voyages encounter exactly the same conditions of weather on any two trips.

From the foregoing multiplicity of variable quantities entering into the problem, it is readily seen how futile is the attempt to arrive at an exact result in regard to the coal required for any one voyage.

The intention of this paper is to indicate a means of making the best of the situation that confronts a commander of a ship or of a fleet on any particular occasion. The object may be to make as long a leg on a voyage as practicable, or to make the greatest possible speed to a certain port and have sufficient coal left in the bunkers to carry on operations after arrival.

The method includes a table derived from day-to-day experience on the particular voyage with all the variable features entering into the problem. At the end of a day or any period of time, called the constant period, in which the conditions noted above are considered practically constant, a numerical factor is deduced which may be called the "*Factor of assurance.*" It indicates at a glance just what has been done and what may be expected under average conditions of the constant periods, and enables the captain or the commander in

chief to increase or decrease speed a definite amount according to the object he has in view.

The *factor of assurance* is unity when the probability of reaching the destination with a certain predetermined amount of coal on hand is assured. If it is desired to reach the destination in the shortest possible time, the object of the captain or the commander in chief will be to regulate the speed so as to keep the *factor of assurance* as near unity as possible. This can be done by increasing or decreasing speed from time to time.

If it is desired to reach safely a destination, which approaches the radius of action of the ship, without regard to the time spent on the voyage, the object will be to regulate the speed so as to keep the *factor of assurance* an amount above unity, which in his judgment will be safe. In this case, if the ship steams at her most economical speed, the *factor of assurance* will increase rapidly toward the end of the voyage until it reaches infinity at the destination, and thus describe a hyberbolic curve. In the former case it should describe a straight line.

The table for calculating the *factor of assurance* should contain columns as follows:

1. Date and hour of completion of constant period.
2. Speed for which revolutions are made.
3. (*a*) Distance run during constant period.
4. (*b*) Coal consumed during same.
5. (*c*) Miles per ton of coal for same.
6. (*d*) Coal on hand.
7. (*e*) Distance ship can steam under same conditions as obtained during preceding constant period.
8. (*f*) Distance to destination.
9. (*F*) Factor of assurance obtained from the above.
10. Conditions of wind and sea and other conditions that affect the speed and coal consumption.

The figures in the various columns are deduced as follows:

$$c = \frac{a}{b} \qquad e = c \times d \qquad F = \frac{e}{f}$$

The following table, showing an actual performance, is inserted as an illustration of the problem pertaining to an individual ship.

TABLE OF FACTORS OF ASSURANCE OF U. S. S. "BENNINGTON" ON RUN FROM HONOLULU TO SAN DIEGO, CAL.

| Date (noon). | Speed for which revolutions are made. | (*a*) Distance run last 24 hours. | (*b*) Coal consumed last 24 hours. | (*c*) Miles per ton last 24 hours | (*d*) Coal on hand. | (*e*) Distance ship can steam with coal on hand. | (*f*) Distance to destination. | (*F*) Factor of assurance. | Conditions of wind and sea last 24 hours. |
|---|---|---|---|---|---|---|---|---|---|
| July 10.. | ... | ... | ... | ... | 304.00 | ... | 2,125 | ... | In port. |
| July 11.. | 9.0 | 221 | 24.73 | 8.94 | 279.27 | 2,496.6 | 1,904 | 1.31 | Light breeze ahead, sea smooth. |
| July 12.. | 9.0 | 216 | 25.80 | 8.37 | 253.47 | 2,121.5 | 1,688 | 1.25 | Same as before until 8 A. M., then moderate sea ahead. |
| July 13.. | 9.0 | 190 | 26.00 | 7.30 | 227.47 | 1,660.5 | 1,498 | 1.10 | Moderate breeze and sea ahead. |
| July 14.. | 9.0 | 222 | 24.2 | 9.10 | 203.27 | 1,850.0 | 1,276 | 1 45 | Light breeze ahead, sea smooth. |
| July 15.. | 10.5 | 252 | 27.8 | 9.00 | 175.47 | 1,579.0 | 1,024 | 1.54 | Same as before. |
| July 16.. | 11.5 | 280 | 31.5 | 8.88 | 143.97 | 1,278.0 | 744 | 1.73 | Do. |
| July 17.. | 12.5 | 284 | 36.3 | 7.80 | 107.67 | 839.0 | 460 | 1 82 | Moderate breeze, moderate head sea. |
| July 18.. | 11.0 | 268 | 28.3 | 9.40 | 79.37 | 746.0 | 192 | 3.88 | Sea smooth, light airs. |
| July 19.. | 10.0 | 192 | 21.0 | 9.15 | 58.37 | 534.0 | ... | ... | Same as before |

Upon leaving Honolulu, July 10, the speed was set at 9 knots, and the number of revolutions ordinarily required to make this speed was prescribed. At noon, July 11, $F$ was found to be 1.31. This being considered good, no change was made in the revolutions. July 12 $F$ was found to be 1.25. This was also considered good, and no change was made, although a moderate head sea was encountered at 8 A. M. July 13 $F$ was 1.10. A moderate head sea and breeze had been encountered throughout the preceding constant period. The ship pitched considerably and kept the bridge wet with spray. Had this weather continued for the remainder of the voyage there would have been a margin of safety for the ship of $1{,}498 \times 0.10 = 149.8$ miles, and the coal remaining after arrival would have been $227 \times 0.10 = 22.7$ tons. Although this margin was rather narrow, the revolutions were unchanged. Besides, there were indications of the weather moderating. July 14 $F$ was 1.45. The weather had moderated soon after noon of the preceding day. Advantage was

taken of this to shove the ship along by increasing speed to 10.5 knots.

From then on $F$ continued to increase rapidly, permitting the speed to be increased accordingly. Throughout the voyage there was no apparent change in the efficiency of the machinery; consequently, the conditions of wind and sea only had to be considered.

The same principles illustrated in the foregoing table and remarks may be conveniently made use of by the commander in chief of a fleet or squadron. In this case he would consider the least efficient ship in steaming, or the one which would develop the least efficiency after the voyage began. This would be indicated by the one that gave the lowest value of $F$. If bound on a long cruise, for instance, from Hampton Roads to the Orient via the Straits of Magellan, the determination daily of this factor for each vessel of the fleet will enable him readily to select the stopping places for coal with a view to lengthening the legs of the voyage as much as possible, thereby reducing the number of stops, and consequently shortening the time to his ultimate destination.

Where the saving of coal is not an object the greatest practicable speed can be made between coaling ports, thus still further shortening the time to his ultimate destination.

The amount of coal required by each ship at the next coaling port can be estimated at a glance after receiving the noon reports.

It may sometime be necessary for the fleet to proceed at the greatest possible speed to a place and there fight a battle before coaling. In this case the ship with the least radius of action would be considered. Suppose that the bunker capacity of such a ship were 2,000 tons and that, in the judgment of the commander in chief, she should have 600 tons on hand when arriving upon the scene of operations. This amount is then reckoned as a reserve. The remainder, 1,400 tons, is the amount allowed for the run, and is so used in the determination of $F$. If during the run the speed of the fleet is so regulated that $F$ will remain at unity, the best possible speed will

be made to the destination, and upon arrival the ship under consideration will have 600 tons on hand, or the required amount.

The same principle may be made use of in so regulating the speed of the fleet and consequent consumption of coal that it may reach the enemy with the best trim of water-line armor for going into action.

The following table for the information of the commander in chief may be made out each day by an aid or by the fleet engineer after receiving the coal reports. This table, called the *fleet table of factors of assurance,* contains columns giving the following information:

1. Date and hour of completion of *constant period.*
2. Standard speed during same.
3. Distance run during same.
4. Distance to destination.
5. A column for each ship showing her factor of assurance.
6. Conditions of wind and sea and other conditions affecting speed and coal consumption.

With this table before him the commander in chief can tell at a glance what each ship is doing in regard to coal consumption, and can readily regulate the speed to suit any conditions he desires to fulfill.

In addition to the uses of this factor, as described above, it is believed that a record of it, combined with other considerations, may be useful in determining the relative steaming efficiency of the various vessels of the fleet.

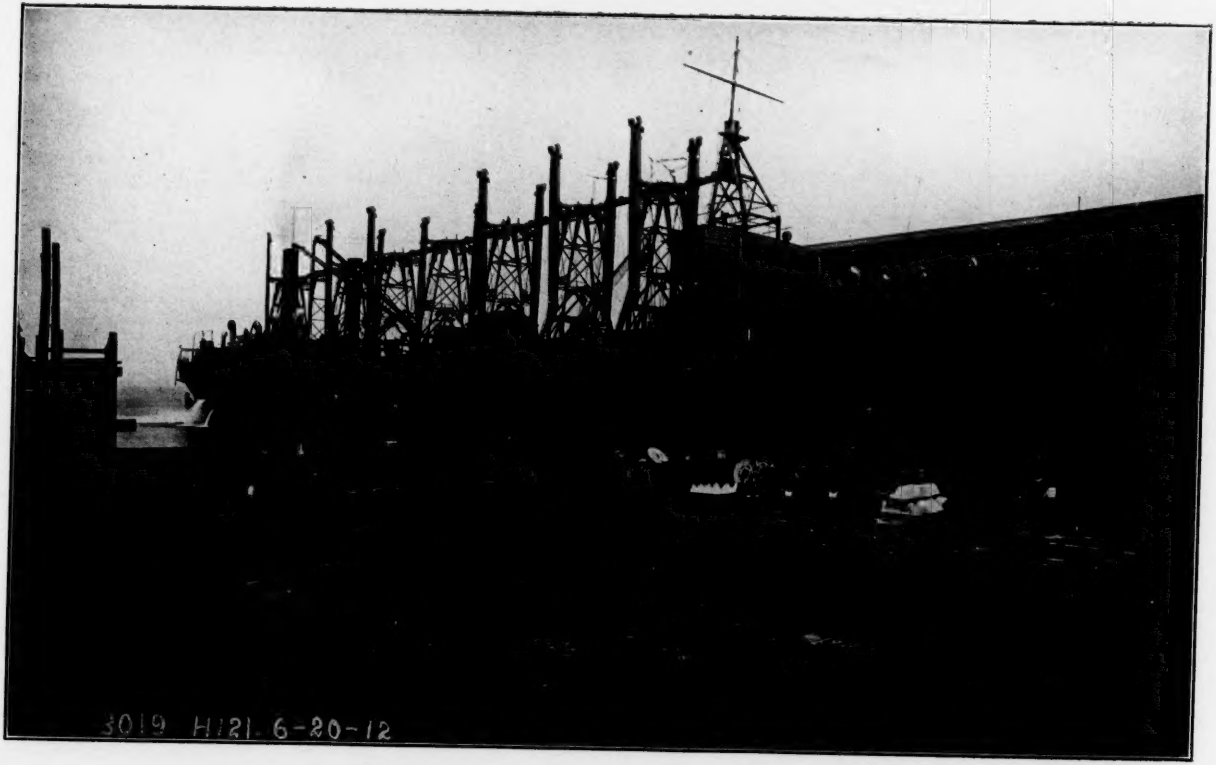



PLATE I.—U. S. S. "ORION" AT BUILDER'S DOCK.

# U. S. FLEET COLLIER *ORION*.

## DESCRIPTION AND TRIALS.

By Lieut. R. L. Irvine, U. S. N., Member.

Fleet Collier No. 11, the *Orion*, was authorized by an Act of Congress of March 4, 1911. The contract was awarded to the Maryland Steel Company, of Sparrow's Point, Md., and was signed August 22, 1911, the price being $951,000.00, and the time of construction, twenty-four months. She was designed for a speed of 14 knots, at about 19,130 tons displacement, with the main engines developing about 6,800 I.H.P.

The keel was laid October 6, 1911, and the vessel was launched March 23, 1912, five months and seventeen days later. On May 30, 1912, a full-power dock trial was held, and four weeks later, June 27, a successful builder's trial was conducted in Chesapeake Bay, followed early in July by the contract trials, the vessel being delivered to the Government and preliminarily accepted on July 24, 1912, at the Navy Yard, Norfolk, Virginia, twelve months and twenty-nine days ahead of the contract time.

The construction of the vessel was in conformity with the Rules of the American Bureau of Shipping and the Rules of the United States Steamboat Inspection Service. Inspection was carried out during the construction of the vessel by the representatives of both organizations and the inspection certificates of both were furnished the Navy Department upon completion of the vessel.

### GENERAL DESCRIPTION.

The *Orion* is a steel, twin-screw, self-trimming, topside-ballast collier. There are three decks forward and three aft, with the forecastle and poop one deck higher than the main deck.

Amidships, in the cargo-bunker section, there is but one deck. There are ten watertight, three oiltight and two non-watertight athwartship bulkheads. The double bottoms are four feet six inches deep and extend from the engine-room after bulkhead to the forward end of the forward cargo hold. The hull is constructed on the "Isherwood" system of longitudinal framing, as shown in Plates 2 and 3, with the shell plating in sunken and raised strakes. The keel is of the flat-plate type. The main, berth and lower decks are of steel, and the forecastle and poop decks of wood with stringer and tie plates of steel. Steel bulwarks, four feet and three inches high, are built in the waist.

A Hyde anchor windlass with two gipsy heads is located on the forecastle, the anchor engine being on the main deck under the windlass. The vessel is fitted with patent anchors which stow in the hawse pipes. The lamp, paint and oil room is in the bow forward of the anchor engine. Below the main deck forward are small store rooms, chain lockers, and a peak tank.

Between the forecastle and bridge are four cargo oil-fuel deep tanks, Nos. 1, 2, 3 and 4, connected to each other by sluice valves and having connections with the fuel-oil pump in the engine room for loading and discharging the oil; and with the ballast pump, also in the engine room, for admitting or discharging water ballast. There is also a connection from the bottom of each tank for removing water which may settle from the cargo oil.

Well forward there are a flying bridge and a lower bridge. On the flying bridge are located the steering wheel, compass, a double-faced engine-room telegraph, engine-room bell pulls and voice tube, docking telegraph, steering telegraph, a 24-inch searchlight, and leadsmen's platforms. The lower bridge is protected on the forward side by a steel bulkhead with large air ports. On the starboard wing of this bridge is a large emergency cabin and a toilet, and on the port wing is a spacious chart house. In the midship space on this bridge are located a compass, a double-faced engine-room telegraph, en-

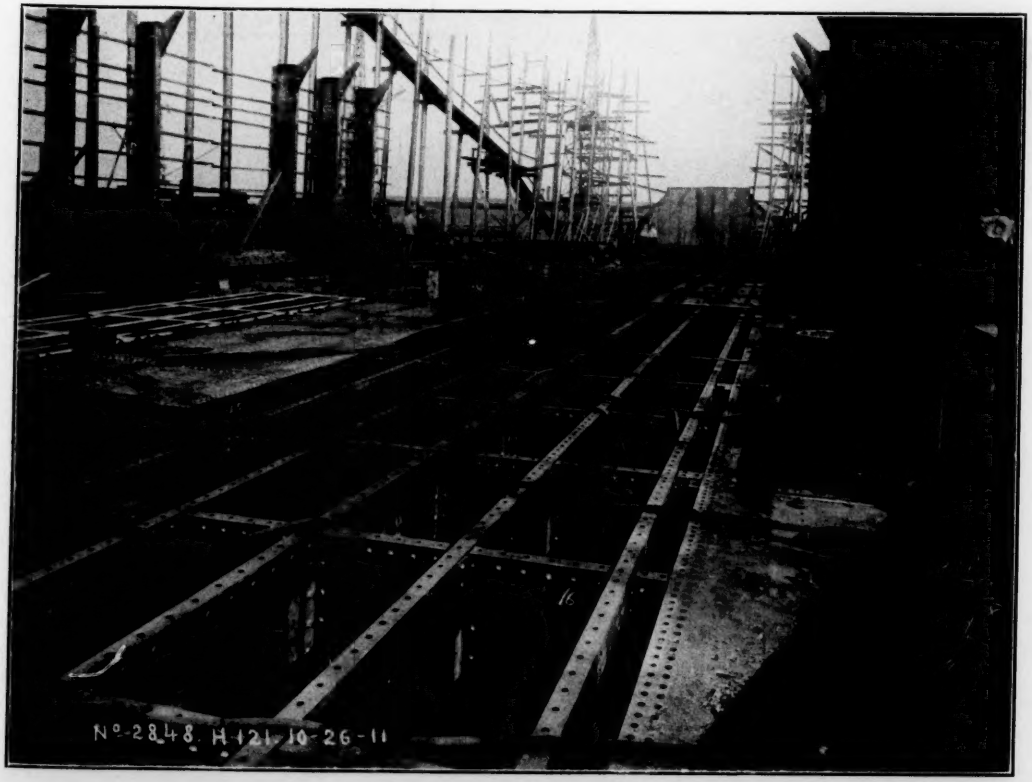



PLATE II.—U. S. S. "ORION." ISHERWOOD SYSTEM OF FRAMING IN DOUBLE BOTTOMS.

PLATE III.—U. S. S. "ORION," ISHERWOOD SYSTEM OF FRAMING. VIEW IN CARGO HOLD, LOOKING ATHWARTSHIPS.

gine-room bell pulls and voice tube, docking telegraph, steering telegraph and steering wheel with telemotor control.

The cargo coal holds and topside trimming tanks extend from a watertight bulkhead under the bridge to the watertight bulkhead at the forward end of the poop. These coal holds are six in number and extend the full width of the vessel and from the inner bottom to the topside ballast tanks and main deck. Access to them is had by hatches 32 feet by 12 feet 6 inches, there being one such hatch over the forward hold and two over each of the other five.

The double bottoms below the cargo coal holds are made oil-tight and may be used for carrying fuel oil or water ballast, being connected to both the ballast and fuel-oil pumps. The topside trimming tanks are just under the outboard sides of the main deck and outboard of the cargo coal-bunker hatches. They are filled by means of the ballast pump and drain overboard by gravity.

On the poop deck are the forward steaming bunker hatch; the superstructure; and after poop deck, where there is a steam capstan, a towing rail, bitts and a hatch to the main deck.

Within the superstructure from forward, aft, are the bakery and officers' and crew's galleys, located between the two boiler hatches, through each of which a smoke pipe rises, the two smoke pipes being in line athwartship; radio office, passages, after steaming bunker hatch, and warrant officers' state rooms; warrant officers' mess room, located between two passages, outboard of which are the engine-room hatches; officers' toilets, and bath rooms; wardroom country; captain's cabin, dining room and bath; offices, etc

The principal berthing space for the crew is on the main and berth decks aft, below the superstructure.

There are accommodations for fourteen officers, eleven petty officers, and one hundred and fifty-two crew.

The steering engine is on the main deck below the after capstan. Immediately abaft the after cargo coal hold and separated therefrom by a watertight bulkhead, are the main steaming bunker of 1,613 tons capacity, an upper bunker of 506 tons

capacity, and the boiler room with three double-ended Scotch boilers and one single-ended Scotch donkey boiler. On each side abaft the boiler room is a steaming bunker feeding leg. A broad passageway leads to the engine room, abaft of which are peak tanks, fresh-water tanks and cold-storage plant.

The inside of tanks and double bottoms where it is intended to carry oil are not painted but are coated with crude oil. Fresh-water tanks are given a cement wash inside. Bitumastic cement, bitumastic solution, and bitumastic enamel are used as a protection to the interiors of coal bunkers and peak and ballast tanks.

The general dimensions of the vessel are as follows:

| | |
|---|---|
| Length over all, feet and inches | 536–0 |
| between perpendiculars, feet and inches | 514–0 |
| on load-water line, feet and inches | 514–0 |
| Beam, extreme, feet and inches | 65–2½ |
| on load-water line, feet and inches | 65–2½ |
| moulded, feet and inches | 65–0 |
| Mean designed draught, feet and inches | 27–4½ |
| Displacement on designed draught, tons | 19,130 |
| per inch on load-water line, tons | 64 |
| Area of immersed midship section at designed draught, square feet | 1,753 |
| of load-water line plane at designed draught, square feet | 26,728 |
| Wetted surface at designed draught, square feet | 46,960 |
| Coefficient of fineness, block | 0.726 |
| midship section | 0.986 |
| load-water line | 0.800 |

## COAL HANDLING GEAR.

Extending fore and aft amidships from the foremast to the main mast and plumbing the cargo and steaming bunkers is a horizontal track for transferring coal fore and aft. This track is in portable sections supported by seven athwartships "A" frames and the bridge. These "A" frames are about 56 feet high above the main deck, and all except the after one, which is on the poop and forms the base of the main mast, are built up from the main deck. A small supporting frame is worked in between the smoke pipes. These frames also form the main support for the booms, stays and running gear used

in loading and discharging cargo and bunker coal. Pivoted by goose-neck joints near the base of each side of each "A" frame, except the after one, are two steel-girder booms of box section, suitably guyed, by means of which the coal conveyors are directed and the cables supported. The after "A" frame has but one such boom on each side, that being all that are necessary for serving the after steaming bunker hatch.

There are twenty-four coaling winches of the double-drum Lidgerwood type having single gipsy heads and double nine-inch by ten-inch steam cylinders. Each winch is fitted with a sheet-metal guard close up under the drums to prevent the cable from overriding the ends of the drums. There are four of these winches under each of the forward five "A" frames, and two under each of the after two, thus giving one bucket winch and one swinging winch for each hatch, including the after steaming-bunker hatch.

The coal is handled by mechanically-operated clamshell buckets of the Hayward Class E type, each having a capacity of 1⅛ cubic yards. When not in use these clamshells are stowed in chocks on the main deck abreast the cargo coal hatches.

The contractors guaranteed a discharging capacity of 100 tons per hour per cargo hatch, making a total of 11,000 tons per hour when discharging from all hatches simultaneously. This guarantee was fully met on trials conducted on July 12, 1912, at the Navy Yard, Norfolk, Virginia, in the presence of members of the official trial board.

## DESCRIPTION OF MACHINERY.

*Main Engines.*—The main propelling machinery consists of two vertical, inverted, direct-acting, three-cylinder, triple-expansion engines, each having a stroke of 48 inches, and with the following cylinder diameters: H.P., 27 inches; I.P., 46 inches; and L.P., 76 inches. The engines are outboard-turning when running ahead. They are located abreast of each other, with all the principal auxiliary machin-

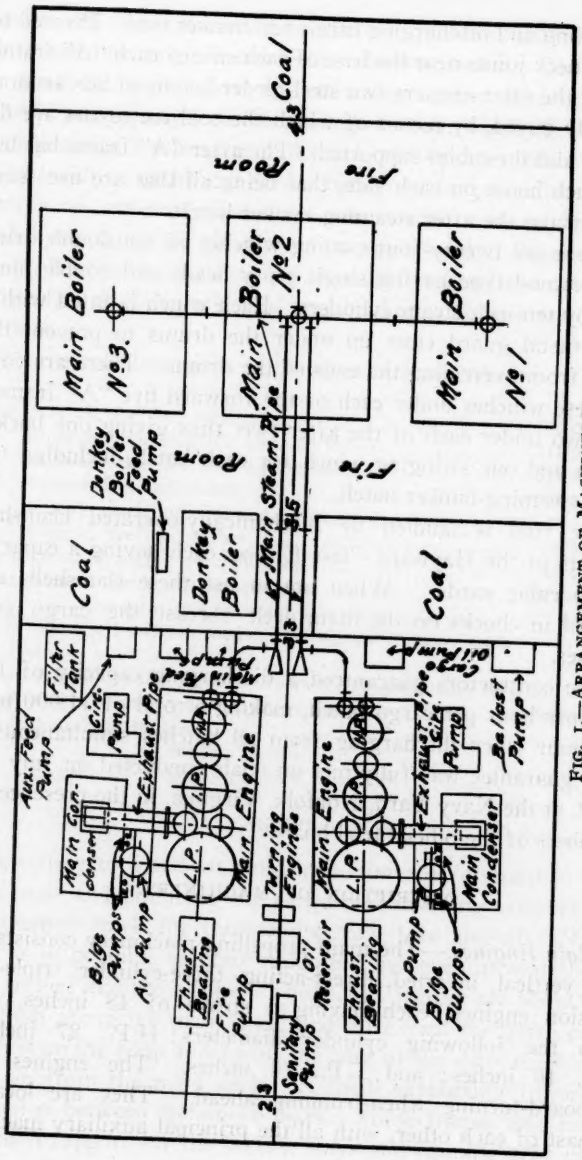



FIG. I.—ARRANGEMENT OF MACHINERY.

ery in one large compartment, as shown in Figure 1. They are designed to develop about 6,800 horsepower when running at 90 revolutions per minute, with steam pressure at 195 pounds at the H.P. valve chest. All the valves are of the piston type and are fitted with balance pistons. Valves and valve settings are as per following table:

| | *H.P.* | | *I.P* | | *L P.* | |
|---|---|---|---|---|---|---|
| Number | One. | | One. | | Two. | |
| Diameter, inches | 13½ | | 25 | | 25 | |
| Travel, inches | 9 | | 9 | | 9 | |
| Side on which steam is taken | Inside. | | Outside. | | Outside. | |
| | *Top.* | *Bottom.* | *Top.* | *Bottom.* | *Top.* | *Bottom.* |
| Width of port, inches | 3 | 3 | 3½ | 3½ | 3½ | 3½ |
| Steam opening, linear, inches | $2\frac{5}{32}$ | $2\frac{11}{32}$ | $2\frac{7}{32}$ | $2\frac{15}{32}$ | $2\frac{5}{16}$ | $2\frac{9}{16}$ |
| Steam opening, area, square inches | 60.8 | | 132 | | 290 | |
| Exhaust opening, linear, inches | 3 | 3 | 3½ | 3½ | 3½ | 3½ |
| Exhaust opening, area, square inches | 86.25 | | 205 | | 412 | |
| Steam lap, inches | $2\frac{1}{32}$ | $2\frac{5}{32}$ | $2\frac{9}{32}$ | $2\frac{1}{32}$ | $2\frac{5}{16}$ | $1\frac{15}{16}$ |
| Exhaust lap, inches | $-\frac{1}{4}$ | $+\frac{1}{8}$ | $-\frac{3}{8}$ | $+\frac{3}{32}$ | $-\frac{1}{4}$ | $+\frac{1}{4}$ |
| Steam lead, linear, inches | $\frac{5}{16}$ | $\frac{1}{4}$ | $\frac{3}{8}$ | $\frac{3}{8}$ | $\frac{1}{4}$ | $\frac{1}{4}$ |
| Cut-off, in decimal of stroke | .730 | .666 | .7415 | .681 | .7502 | .6912 |

Connecting rod, length between centers, inches, 111. Ratio to crank, 4.625.

The engine columns are of cast iron, the front ones being of box type and the back ones of inverted Y type. An innovation in naval practice is introduced here. The front columns were cast with a web closing the bottoms, the insides were well cleaned of sand and scale, pickled, and fitted for the storage of lubricating oil. They have a total capacity of about 1,500 gallons. Cast-iron bar guides, cored for water circulation, are fitted to the back columns and have given entire satisfaction in the trials held thus far.

The oiling system for the main engines is wick feed from brass manifold oil boxes. Forced lubrication is supplied to the thrust blocks by means of two vertical, simplex oil pumps driven by the driving beams of the main air pumps.

Standard United States metallic packing is used on all main engine piston rods and valve stems.

The throttle and reversing lever are located at the forward side of the forward front column on each main engine valve. The throttle is an 8-inch valve of the double-disc type with

body and spool of cast iron, valve seat of hard bronze, and valve stem of Tobin bronze. By-passes are fitted for live steam to the I.P. and L.P. valve chests.

*Reversing Gear.*—The reversing engine is secured to the outboard side of the main engine, abreast of the I.P. valve chest and cylinder. It is of the ram type, with a 26-inch stroke and a cylinder 15 inches in diameter. It is operated from the working station through the reversing lever and a floating lever operated by the reverse shaft. The rocker arms are slotted and fitted with screw blocks for independent linking up while the engine is running.

*Jacking Gear.*—For jacking over the main engines two vertical, single, reciprocating engines are located amidships in line with the after ends of the main L.P. cylinders. Both take steam from the same throttle and act on the same turning shaft, which is fitted with a clutch for each main engine. Each engine has a 6-inch stroke and a 6-inch steam cylinder.

*Bed Plate.*—The main-engine bed plate is of cast iron of box form, flanged and well bolted to the engine foundation with cast-iron liner blocks accurately fitted in way of each holding-down bolt. The bottom main bearing boxes are of cast iron, semi-circular in form, cored for water circulation, and fitting into square bottom saddles which fit recesses in the bed plate. The top box and binder is made in one piece of cast steel; both boxes are lined with Shonberg's bearing metal.

*Shafting and Bearings.*—All shafting is forged steel of solid section.

| | |
|---|---|
| Crank shaft, length, feet and inches | 23–07½ |
| diameter, inches | 14¾ |
| pins, length, inches | 16 |
| diameter, inches | 15 |
| Thrust shaft, length, feet and inches | 15–05¾ |
| diameters, inches | 14¾ and 15 |
| Line shaft | No line shafting. |
| Stern-tube shaft, length, feet and inches | 26–04 |
| diameter, inches | 15⅝ |
| Propeller shaft, length, feet and inches | 35–06 |
| diameter, inches | 15⅝ |

All flanged couplings of shafts are forged on and secured together by taper-fitted forged-steel bolts. The crank shafts are built up, in two sections, one section for the L.P. and one section for the I.P. and L.P. cylinders, with the three cranks at equal angles. The webs are rough-finished only in accordance with merchant practice. The tail shaft is covered through the stern tube with composition sleeves ¾ inch thick through bearings and ½ inch thick between bearings. The sleeves are shrunk and pinned on, with joints between the sections soldered. The shafting is arranged so that the tail shaft can be readily withdrawn.

| | |
|---|---|
| Crank-shaft bearings, number of | 6 |
| length, inches | 17¼ |
| Thrust-shaft bearings (2), length, inches | 16 |
| Collars on thrust shaft: | |
| Number | 7 |
| Thickness, inches | 2⅝ |
| Space between, inches | 5¼ |
| Outside diameter, inches | 24½ |
| Inside diameter, inches | 15 |
| Bearing surface, square inches | 2,063.11 |
| Thrust shoes, number | 7 |
| effective (thrust) surface, square inches | 1,355.2 |
| Stern-tube bearings: | |
| Length, forward bearing, inches | 22 |
| after bearing, inches | 57¾ |
| Bearing surface | Lignum vitae. |
| Strut bearings: | |
| Length, inches | 57¾ |
| Bearing surface | Lignum vitae. |

The thrust block is of heavy cast iron fitted with an oil trough. The block rests on a heavy cast-iron sole plate with a long base, well secured to the structure of the ship.

*Propellers.*—There are two true-screw propellers each having three adjustable, detachable blades of manganese-bronze. The hubs are of cast steel and the caps of cast iron. The blades are secured to the hubs by studs of high-grade steel and composition nuts. The propeller has a taper fit on the tail shaft, is securely keyed, and is held on by a steel nut.

Propeller data:

| | |
|---|---|
| Diameter, feet and inches | 16–06 |
| Pitch as set, feet and inches | 18–00 |
| Pitch adjustable from 17 feet to 19 feet. | |
| Ratio of diameter to pitch | 1.091 |
| Projected area, square feet | 69.97 |
| Helicoidal area, square feet | 82.5 |
| Disc area, square feet | 213.82 |
| Height of lower tip of blade above keel, inches | 19 |
| Immersion of upper tip of blade at load draught, inches | 113 |

*Main Condensers.*—There are two independent, cylindrical, surface, double-flow, main condensers located one outboard of each main engine. The shells are of wrought steel, heads of cast iron, tube sheets of rolled brass $1\frac{1}{8}$ inches thick, and tubes of solid-drawn composition No. 16 B.W.G. thick, $\frac{5}{8}$ inch outside diameter and 10 feet $2\frac{1}{4}$ inches long. The ends of the tubes are packed in tube sheets and held in place with screw ferrules. There are 3,259 tubes in each condenser, giving a total cooling surface in each of 5,331.7 square feet. The circulating water is supplied by a double-suction centrifugal pump of ample capacity, having composition runners and shaft, with a 14-inch suction and discharge. Each pump is located just forward of its condenser and is driven by a single, vertical, reciprocating engine having a 10-inch cylinder and a 9-inch stroke.

*Air Pumps.*—The air pumps, one for each main engine, are direct-connected and are located under the condensers and outboard of the main-engine L.P. cylinders. They are of the Edwards type, built by the Maryland Steel Company, and have 22-inch single water cylinders with an 18-inch stroke. The pump body is of cast iron, brass lined and fitted with brass bucket and delivery valves.

*Bilge Pumps.*—There are two direct-connected bilge pumps of the plunger type for each main engine.

*Auxiliary Condenser.*—A small auxiliary condenser of the Wheeler (combined) surface type is located on the dynamo flat abaft the starboard main engine. This pump has a 10-inch steam cylinder, a 12-inch water cylinder and a 12-inch stroke.

*Feed Pumps.*—Two main and one auxiliary feed pumps are

located near the forward engine-room bulkhead on the port side. These are all Warren, vertical, simplex pumps with 14-inch steam cylinders, 9-inch water cylinders, and a 24-inch stroke. There is also one Warren, horizontal, simplex donkey-boiler feed pump located in the recess to port of the donkey boiler. It has a 7-inch steam cylinder, a 4-inch water cylinder and a 7-inch stroke.

*Feed-Water Heater.*—In the engine room, outboard of the main feed pumps, is a 48-inch Reilly multicoil feed-water heater, so arranged that the feed water can be passed through or around it. Between the feed pumps and the feed heater there is a Blackburn-Smith 5-inch feed-water filter and grease extractor, through or around which the feed water can be passed.

*Cargo-Oil Pump.*—The cargo-oil pump is located at the engine-room bulkhead just forward of the starboard main engine. It is a Warren, vertical, duplex pump having 12-inch steam cylinders, 14-inch oil cylinders, and a 12-inch stroke. It is fitted with metallic valves.

*Ballast Pump.*—The ballast pump is located at the forward engine-room bulkhead, outboard of the cargo-oil pump. It is an exact duplicate of the cargo-oil pump, and the ballast and cargo-oil pipe lines are cross-connected.

*Fire Pump.*—The fire pump is located amidship in the after part of the engine room. It is a Warren. vertical, duplex pump with 10-inch steam cylinder, 7½-inch water cylinders and a 12-inch stroke.

*Sanitary Pump.*—A Warren, vertical, simplex sanitary pump is located in the after part of the engine room to starboard of the fire pump. It has a 6-inch steam cylinder, a 7-inch water cylinder and an 8-inch stroke.

*Lubricating-Oil Pump.*—Attached to each main air-pump beam is a vertical, simplex oil pump for supplying forced lubrication to the thrust blocks. It has a 2-inch oil cylinder and a 10-inch stroke.

*Boilers.*—There are three main, horizontal, double-ended Scotch boilers and one horizontal, single-ended Scotch donkey boiler, all fitted with separate combustion chambers. The

main boilers are placed abreast of each other athwartships, with two wide fore-and-aft passages between them. The donkey boiler is located in a recess in the after end of the boiler compartment, between the two after legs of the main steaming bunker and to port of the passageway to the engine room.

| Boiler data: | *Main (each).* | *Donkey.* |
|---|---|---|
| Designed working pressure, pounds | 200 | 200 |
| Test pressure, pounds | 300 | 300 |
| Ratio of G. S. to H. S | 1 to 43 | 1 to 28.3 |
| Length of grates, feet and inches | 5–06 | 6–00 |
| Width of grates, inches | 40 | 44 |
| Per cent. of air space in grates | 39.4 | 40 |
| Diameter of boiler, feet and inches | 16–01 $^{9}/_{16}$ | 8–01⅝ |
| Length, external, feet and inches | 21–04⅝ | 10–06 |
| Number of furnaces | 8 | 1 |
| Grate surface, square feet | 146.7 | 22 |
| Heating surface, square feet | 6,307 | 622.7 |
| Kind of tubes | Charcoal iron, lap welded. | |
| Ordinary tubes, No. 10 B.W.G | 620 | 39 |
| Stay tubes, No. 7 B.W.G | 400 | 44 |
| Screw-stay tubes, ¼-inch thick | 44 | 00 |
| Outside diameter of tubes, inches | 02½ | 03 |
| Length of tubes as fitted, feet and inches | 7–07⅝ | 7–11¾ |

The boilers were built strictly according to the U. S. Steamboat Inspection Rules. The furnaces are of the Morison (suspension corrugated) type and are not fitted with ash pans. Under each front connection door a small door is fitted for removing collections of soot without the necessity of opening the front connection doors.

The furnace fittings consist of cast-iron front and door protected by cast-iron liners, cast-iron dead plates and bridge plate, and double grate bars, in two lengths, arranged for burning soft coal. Ash-pit dampers and air valves, operated at the furnace doors, are fitted for the forced-draft system. Cast-iron plates, covered with fire brick, extend from the top of the bridge-wall at the back to the back side of the combustion chamber. The mountings on each boiler consist of—

1 main steam stop valve,
1 auxiliary stop valve (large enough so that one boiler will run all auxiliary machinery on deck and in the engine-room),
1 nickel-seated twin pop safety valve with easing gear,
1 main feed stop and check valve,
1 auxiliary feed stop and check valve,
1 surface-blow valve,
1 bottom-blow valve,
1 drain cock,
1 water column,
Salinometer,
Try cocks,
Zinc plates in baskets.

The boilers are covered down to the saddles with 1½ inches block magnesia, ½ inch plaster and hard finish, and No. 18 galvanized iron.

*Smoke Pipes.*—There are two smoke pipes, each having a height of 93 feet 6 inches above the grates and an area of 37.8 square feet. The uptakes from the starboard main boiler and the forward end of the middle main boiler connect with the starboard smoke pipe; and the uptakes from the port main boiler, the after end of the middle main boiler, and the donkey boiler connect with the port smoke pipe.

*Forced Draft.*—Howden, closed ash pit, heated air, forced draft is used on the boilers. The air is supplied by two vertical, engine-driven, Sturtevant No. 9, double-inlet, multivane-fan blowers located over the donkey boiler. Each blower engine has a 5-inch steam cylinder and a 4-inch stroke. The air is drawn from the firerooms and delivered to the boilers through sheet-iron ducts of rectangular section, being heated in the boiler uptakes on its way to the ash pits.

*Ash Hoists.*—A hand ash hoist is fitted in the ventilator in the forward fireroom and an hydraulic ash ejector, made by the Maryland Steel Company, is fitted in each fireroom, the forward one discharging to port and the after one to starboard.

*Steam Lines.*—A 7-inch main steam pipe of seamless steel .28 inch thick leads from each main boiler stop valve to two 8-inch main steam pipes of seamless steel, .3 inch thick, cross-connected over the middle boiler, and leading aft through the engine-room bulkhead, from whence one line leads to each main-engine throttle valve. An auxiliary steam pipe of seamless steel 3 inches in diameter and .24 inch thick is taken from each main boiler and all unite into the auxiliary steam line, which is also 3 inches in diameter and .24 inch thick. The steam from the donkey boiler is taken through a seamless steel pipe 3 inches in diameter and .2 inch thick, leading to the auxiliary steam line, which it joins over the middle main boiler. The auxiliary steam pipes beyond reducing valves are of copper. The flanges of all steel steam pipes are rolled. Many joints in the copper auxiliary steam and exhaust pipes and in oil and water piping have screwed fittings, according to merchant practice.

*Ballast and Bilge System.*—The ballast pump has an 8-inch discharge to deck, branching into two 6-inch pipes, one along each side of the main deck close in to the cargo hatches. Each of these lines has branches and valves for filling the topside ballast tanks, which empty by gravity through a valve in the bottom of each tank operated from the weather deck. Branches for the fire main are taken off this line. The forward and after trimming tanks have connections to this pump and also to the bilge manifold. The main-engine bilge pumps draw from the bilge manifold as well as direct from the engine-room bilge. From the manifold suction, pipes lead to the well in the after end of the engine room and two pipes lead to the after end of each main cargo hold. The bilge pipes are of galvanized iron, 3 inches in diameter, and end in strainer boxes. The bilge manifold is of cast iron and a McComb strainer is fitted in the pump suction. Each cargo-oil deep tank has a 2-inch pipe connection in the bottom for draining off any water which may settle from the oil.

*Cargo-Oil Pipes.*—A galvanized-iron pipe 5 inches in diameter leads from the center, and a 3½-inch galvanized-iron pipe

from each wing, of each double-bottom cargo-oil tank, except where the ends fine up, where only the 5-inch pipe is provided. From each cargo-oil deep tank a 5-inch galvanized-iron pipe is taken. These all lead to the cargo-oil manifold at the cargo-oil pump in the forward end of the engine room. The oil pump discharges to the main deck in the waist, where the lines lead to two stand pipes on each side for discharging fuel oil wherever desired by means of portable hose, or for receiving oil on board from a supply alongside. The pump is by-passed so that the suction can be taken from these deck stand pipes and discharged to the tanks; or the tanks can be filled direct from the deck line by gravity. Air and sounding pipes are fitted to all tanks, and graduations for depth are marked on a bulkhead in each cargo-oil deep tank.

*Evaporating and Distilling Plant.*—Two Reilly multicoil evaporators, each having a capacity of 4,000 gallons of fresh water per 24 hours, and two Reilly multicoil cast-iron shell distillers, each having a capacity of 2,000 gallons per 24 hours, are fitted on the port side of the dynamo flat in the after part of the engine room.

One Warren, vertical, simplex evaporator feed pump is located near the evaporators. It has a 5½-inch steam cylinder, a 5-inch water cylinder and an 8-inch stroke.

One Warren, vertical, simplex distiller circulating pump is fitted on the port side of the dynamo flat. It has a 6-inch steam cylinder, a 7-inch water cylinder and an 8-inch stroke.

The fresh-water pump is located on the dynamo flat at the starboard after side near the auxiliary condenser. It is a Warren, vertical, duplex pump having 6-inch steam cylinders, 4½-inch water cylinders and a 7-inch stroke.

*Refrigerating Plant.*—The ice-machine room is just abaft the engine room on the port side and is entered from the dynamo flat. It contains one 2-ton refrigerating machine of the ammonia type, manufactured by the Brunswick Refrigerating Company of Brunswick, N. J. The compressor is driven by a small vertical engine. The oil separator, ammonia re-

ceiver and condenser are overhead in the after end of the ice-machine room. The refrigerating rooms are abaft the ice-machine room on the port side of the center line of the ship. There are two large meat compartments located, one aft with 285 feet of pipe, and one outboard with 240 feet of pipe.

Inboard are a vestibule, a compartment with 83 feet of pipe for officers, and a compartment with 20 feet of pipe for the crew.

Each room is independently controlled.

*Steam Fire-Extinguisher System.*—A steam fire-extinguisher system is fitted with steam taken from the auxiliary steam line. Manifolds are fitted and an independent branch of 1½-inch wrought-iron pipe is led to each hold and bunker.

*Fresh-Water System.*—Tanks are fitted under the machinery space for fresh water for boiler use. These tanks are fitted with connections from the condensers and feed pumps, filling pipes from deck, and air and sounding pipes. A filter box, taking the discharge from the air pumps, is installed in the port forward corner of the engine room, fitted with ball float and gear to control the feed pumps.

*Lighting Plant.*—Two Sturtevant, 25-kw. vertical, engine-driven, direct-connected, direct-current generators are installed amidships on the dynamo flat for lighting purposes. The switch board, of black enameled slate, is at the bulkhead just abaft the generators.

Current is supplied at 125 volts, with a drop of not over 3½ volts at the most distant light when all lights of the circuit are burning. The vessel is wired for 350 lights of 16-candle power each and twenty-five 12-inch desk fans of ¼ horsepower each.

A four-circuit running-light telltale board is installed in the pilot house and connected to the side lights, range lights and masthead light.

A radio room is provided, the outfit to be provided and installed by the Government.

*Store Room.*—An engine-room store room is built in on a level with the dynamo flat, in the port side of the engine-

room compartment. It has vertical wooden-slat sides and is fitted inside with lockers and drawers with locks and keys.

*Workshop*.—No workshop is fitted. A work bench is fitted in the after end of the engine room between the shafts, and a vise, anvil, portable forge, and an outfit of hand tools are supplied.

## TRIALS.

The contract required:

(*a*) *A progressive trial* of about 17 runs over the mile course at Delaware Breakwater for the purpose of standardization the screw, with the vessel in full-load condition.

(*b*) A full-speed trial of 48 hours' duration, in the open sea, in deep water, at an average speed of not less than 14 knots. On this trial it was required that the air pressure should not exceed 1¼ inches of water, and the steam pressure at the H.P. valve chests should not exceed 195 pounds above atmospheric pressure.

(*c*) Complete trials of the coaling appliances and apparatus of the vessel. Also similar trials of the oil-fuel handling appliances.

*Standardization Trial*.—On July 9, 1912, the standardization trial was run on the mile course off Delaware Breakwater. In all, seventeen runs were made over the course at various speeds, as shown in Table I, which gives the principal data. The estimated displacement at the middle of the high-speed runs was 19,089 tons, and corresponding mean draught 27 feet 4 inches. The curves, Figure 2, were plotted from the data given in Table I. From the official curves it was found to require 91.4 r.p.m. of the main engines to give the contract speed of 14 knots.

*Forty-eight Hours Full Speed Trial*.—This trial was held in the open sea on July 10th, 11th and 12th, 1912. The weather during the trial was exceptionally fine with a smooth sea and a southerly breeze of maximum force of 4, Beaufort scale. For about one hour the first day a thick fog was encountered.

TABLE I.—STANDARDIZATION TRIAL DATA U. S. COLLIER "ORION."

Course off Delaware Breakwater, July 9, 1912.

| Number of run | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Time on course, minutes | 5 | 5 | 6 | 3 | 4 | 3 | 4 | 3 | 4 | 3 | 4 | 5 | 6 | 6 | 6 | 9 | 6 |
| seconds | 50.23 | 1.45 | 13.7 | 39.5 | 32.45 | 36.5 | 31.7 | 36.53 | 42.73 | 54.9 | 41.53 | 43.17 | 26.35 | 4.5 | 59.27 | 25.9 | 7.4 |
| Revolutions to go 1 knot: | | | | | | | | | | | | | | | | | |
| Starboard engine | 422.55 | 362.7 | 448.2 | 362.05 | 441.95 | 357.45 | 438.9 | 363.4 | 429.3 | 361.75 | 426.1 | 343.8 | 407.45 | 383.25 | 352 | 470.4 | 316.35 |
| Port engine | 404.25 | 346.8 | 428.8 | 362.6 | 442 | 357.6 | 440.75 | 362.9 | 432.1 | 360.6 | 426.9 | 374.15 | 409.8 | 386.7 | 339.35 | 469.1 | 306.85 |
| Average | 413.4 | 355.05 | 438.5 | 362.33 | 441.98 | 357.53 | 439.83 | 363.15 | 430.7 | 361.18 | 426.5 | 358.98 | 408.63 | 384.97 | 345.68 | 469.75 | 311,6 |
| Revolutions per minute: | | | | | | | | | | | | | | | | | |
| Starboard engine | 72.37 | 72.26 | 72.02 | 98.99 | 97.6 | 99.06 | 97.07 | 100.69 | 91.1 | 92.38 | 90.79 | 60.11 | 63.39 | 63.1 | 50.38 | 49 86 | 51.23 |
| Port engine | 69.23 | 69.10 | 68.91 | 99.14 | 97.61 | 99.12 | 97.48 | 100.55 | 91.69 | 92.09 | 90.96 | 65.42 | 63.75 | 63.66 | 48.56 | 49.72 | 50.01 |
| Average | 70.80 | 70.68 | 70.47 | 99.07 | 97.61 | 99.09 | 97.28 | 100.62 | 91.4 | 92.24 | 90.88 | 62.77 | 63.57 | 63.38 | 49.47 | 49.79 | 50.62 |
| Speed in knots | 10.279 | 11.942 | 9 633 | 16.401 | 13.213 | 16.628 | 13.25 | 16.628 | 12.734 | 15.326 | 12.787 | 10.49 | 9.318 | 9.876 | 8.587 | 6.350 | 9.799 |
| I.H.P., starboard engine | 1,597 | 1,492 | 1,471 | 3,609 | 3,920 | 4,068 | 3,869 | 3,809 | 3,191 | 3,156 | 3,125 | 861 | 1,048 | 1,005 | 522 | 586 | 658 |
| port engine | 1,359 | 1,326 | 1,314 | 3,649 | 3,167 | 3,800 | 3,564 | 3,901 | 3,272 | 3,194 | 3,185 | 1,224 | 1,102 | 1,067 | 531 | 534 | 561 |
| Total | 2,956 | 2,818 | 2,785 | 7,258 | 7,087 | 7,868 | 7,433 | 7,710 | 6,463 | 6,350 | 6,310 | 2,085 | 2,150 | 2,072 | 1,053 | 1,120 | 1,219 |

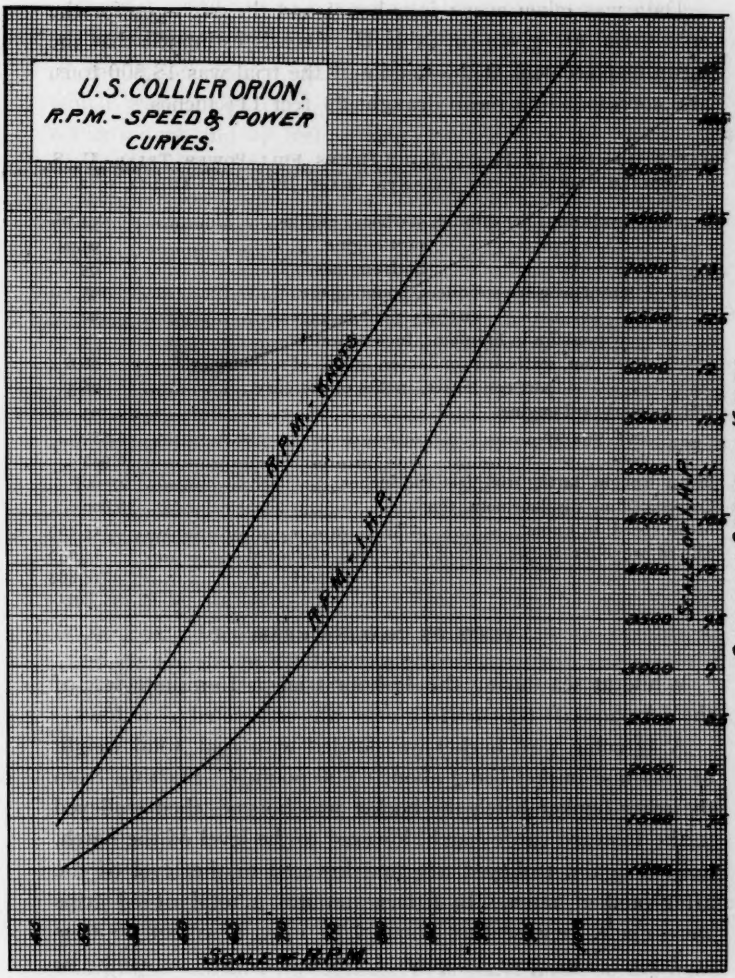

U.S. COLLIER ORION.
R.P.M. - SPEED & POWER CURVES.
R.P.M. - KNOTS
R.P.M. - I.H.P.
SCALE OF R.P.M.
SCALE OF I.H.P.
SCALE OF SPEED IN KNOTS.

FIG. 2.

The trial was entirely satisfactory and the contract speed of 14 knots was easily exceeded.

Data was taken every four hours and the averages for the entire trial are given in Table II. It was estimated that the mean displacement at the middle of the trial was 18,800 tons, and corresponding mean draught 26 feet 11⅞ inches.

TABLE II.—DATA OF FORTY-EIGHT HOURS FULL-POWER TRIAL, U. S. COLLIER "ORION," OFF ATLANTIC COAST, JULY 10, 11 AND 12, 1912.

| | |
|---|---|
| Average revolutions per minute, starboard engine | 94.76 |
| port engine | 95.24 |
| both engines | 95 |
| steam pressure at boilers, pounds gage | 195 |
| engines, pounds gage | 192 |
| air pressure in firerooms, inch of water | 1 |
| vacuum, starboard condenser, inches of mercury | 26.6 |
| port condenser, inches of mercury | 26.5 |
| I.H.P., starboard engine | 3,393 |
| port engine | 3,550 |
| total | 6,943 |
| Mean draught during trial, feet and inches | 27-00⅝ |
| Corresponding displacement, tons | 18,800 |
| Mean speed, in knots | 14.469 |
| Slip of propeller, in per cent. of own speed, starboard | 14.056 |
| port | 14.488 |

## TESTS OF COALING GEAR.

While at sea during the 48-hour trial, a 15-minute test of the fore-and-aft transporting gear was conducted as follows:

A 1⅝-cubic yard bucket was loaded with coal, total weight of coal and bucket being about 6,600 pounds. Starting from the after bunker hatch the bucket was hoisted, then trolleyed to the forward cargo hatch, lowered to the level of the hatch, hoisted again, trolleyed aft, and lowered to the level of the after bunker hatch. In fifteen minutes a total of 5¾ trips were made, representing about 7.15 tons, or about 28 tons an hour. The gear in general worked as well as could be expected. At the start some trouble was experienced by getting riding turns on the hoisting drums. The chief trouble, however, was due to the inability of the operator to see the

forward hatch and tell when the bucket was clear for hoisting or lowering.

On July 12th the coaling gear was tested for capacity of unloading. A lighter was secured on the port side of the collier and at each end uprights were erected from which was suspended a line which was 27 feet above the collier's deck and 54 feet 6 inches outboard of the center of the hatch. The purpose of the line was to enable the Board to estimate the height and the distance outboard of the hatch to which the coal was hoisted before it was dumped. In addition to this the approved plans specify that the height of the bucket when passing the collier's rail shall be at least 24 feet. This height was estimated, it being impracticable to erect a line at this point because it would interfere with the travel of the bucket. The test was conducted for one hour, at the end of which time 123 buckets had been dumped, containing in all $137\frac{130}{240}$ tons. This was well in excess of the contractors' guarantee of 100 tons per hatch per hour, and the trial was, therefore, not carried any further. The gear and its operation were eminently satisfactory.

During the trials the gear was operated by a skilled man, and, in order to derive the most benefit from the gear, it will be necessary that intelligent men be trained in its operation. As there are eleven hatches, the gear is extensive and requires considerable time to rig.

### TESTS OF FUEL-OIL HANDLING GEAR.

The tests of the fuel-oil handling gear were conducted on June 14 and on June 19, 1912.

On the test held June 14 the fuel-oil deep tanks were partly filled with salt water, as no oil was available, and the fuel-oil pump was used with a suction on the deep tanks and discharging overboard through the four stand pipes on the main deck. The tanks were in communication with each other through the sluice valves. The test was conducted for one hour, in which time 84,000 gallons of water were discharged.

During the second test the fuel-oil pump took a suction from the cargo-oil deep tanks and the ballast pump took a suction from the double bottoms Nos. 1 and 2. Salt water was used and the cross-connection between the two suction lines was closed, but both pumps discharged overboard through the same line—the four stand pipes on the main deck. The pumping was continued for one hour, and at the end of that time it was found that the ballast pump had discharged 73,168 gallons and the fuel-oil pump had discharged 79,000 gallons.

# USE OF FUELS IN THE UNITED STATES NAVY.*

By Engineer-in-Chief H. I. Cone, U. S. N., Member.

The fuels used in the Navy are coal, fuel oil and gasoline.

Coal must have certain characteristics to make it suitable for Naval use.

One of the most important strategical requirements of a warship is her ability to steam great distances without recoaling. Another is her ability, in time of need, to make high speeds. As both bunker capacity and boiler power are limited by other features of design, it follows that the coal must have the greatest thermal efficiency obtainable in connection with the other necessary characteristics.

Considerations of weight and space impose certain restrictions upon boiler design. In naval boilers the ratio between cubic foot of combustion space and pounds of fuel consumed per minute is necessarily much less than in other types. The effect may be most readily grasped if expressed in terms of the length of time that a given volume of the gases generated by the burning coal remains in the furnace. The following figures are quoted from a report of the Bureau of Mines of the Interior Department:

| | Fuel per hr. per sq. ft. of grate. | Time each cu. ft. of gas stays in furnace. |
|---|---|---|
| Torpedo-boat boiler, | 40 pounds | .077 seconds |
| Modern locomotive, | 60 pounds | .17 seconds |
| Stationary (Heine), | 25 pounds | .58 seconds |

It is apparent from the foregoing that naval coal must be as low as possible in volatile content. This second characteristic is as important as the first, for it matters little if the coal is

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high in B.t.u's., but is of such chemical composition that too many of them go up the smoke pipe in the form of unconsumed gases and tarry vapors. But it is usually difficult to get a coal producer to admit the inferiority of his product for naval use when a chemist has certified that laboratory analysis shows it to contain as many thermal units as coal that the Navy is buying, but to run perhaps 20 per cent. less in fixed carbon.

High volatile coals are usually very smoky in naval boilers. Smoke not only reveals the location of the fleet but might fatally interfere with accuracy of gunfire.

Amount of ash is a characteristic affecting not only the ultimate thermal efficiency to a greater degree than laboratory analyses may indicate, but also the amount of labor involved in disposing of the refuse. Exhaustion of the firemen may be the determining factor in battle.

Amount and nature of clinkers has the same bearing as ash. Some coals suitable in other respects fuse to the grates to an extent which soon shuts off the air supply; their removal is the most exhausting work a fireman has to do.

Coal should coke readily to avoid loss through the grates, as fires have to be "worked" vigorously when steaming at high powers and during the frequent, sudden and great changes in speed incident to fleet maneuvers.

The final characteristic is cost. It is fortunate for the Navy that the coals which best fulfill all the other characteristics are usually the cheapest in dollars and cents per horsepower developed.

Extended experience under service conditions and numerous special tests show that the semi-bituminous coal from the great fields in Southeastern West Virginia, Western Maryland and Southern Pennsylvania are the American coals which best fulfill these requirements. They comprise the great bulk of all the coal used in our Navy, both at home and on foreign stations. In special cases where it is desirable or necessary for ships remote from Government colliers or Government coal piles to purchase coal, Admiralty Welsh is usually selected if American

coal of the required kind is not obtainable at lesser cost. Welsh coal is not different from ours in its performance in naval boilers, but is much more likely to cause trouble from explosive gases in the bunkers.

Coal equal or superior to the American coals now used is thought to exist in great quantities in Alaska, but it seems questionable whether the fields will be developed sufficiently to supply even a part of the Navy's needs before the time when most of the coal-burning ships will have been written off the Navy's books.

The advantages of oil as compared with coal are:

An evaporation per pound of fuel in the ratio of about 14 to 9, and per square foot of heating surface in about the ratio of 10 to 8.

Fuel can be taken aboard more rapidly, without manual labor, and without interruption to the routine of the ship. The problem of fueling at sea is solved.

Steam for full power can be maintained as readily as for low power. A vessel burning oil is capable of runs at full speed limited in duration only by the supply of fuel. There is no reduction in speed due to dirty fires or to difficulty in trimming coal from remote bunkers, or to exhaustion of the fireroom force.

There are no cinders and the amount of smoke can be controlled.

A considerable reduction in *personnel* is possible.

The weight and space required for boilers is reduced: First, by the reduction in heating surface required, and second, by the shortening of firerooms. Consequent on the reduction in heating surface is a decrease in weight and cost of boilers.

Coal and ash-handling gear is eliminated. This renders unnecessary the piercing of the hull for coal trunks and discharges from the ash expellers or ash ejectors.

The stowage and handling of oil is much easier than of coal and will result in a much cleaner ship with consequent increase in time available for drills.

The mechanical supply of fuel to the boilers gives a prompt

and delicate control of the steam supply, permitting more sudden changes in speed than with coal, which is a tactical advantage.

The nature of fuel oil permits utilization of remote portions of the ship and of constricted spaces for its stowage.

These advantages have long been recognized by the Navy and there have been experiments with liquid fuel dating back as far as 1867. All these experiments have confirmed our belief in the considerable military advantages which will accrue from its use, but until recently it has been impracticable to use it extensively on account of the uncertainty as to the adequacy of its supply and the sufficiency of its distribution among the seaports of the world. We are now assured, however, as regards the supply, that there is sufficient oil on the public lands of the State of California alone to supply all probable naval demands for one hundred years should oil be burned to the exclusion of coal, and of course there is considerable oil in other portions of U. S. territory. The question as to the distribution of oil among the ports from which fuel might be required by our vessels in time of war is one that is well within our power to solve, as from its nature the oil can be transported and stored more easily than can coal. Indeed for the transport of oil in time of war we are already better provided than for coal, there being a large number of tank steamers flying the American flag.

Oil is therefore certain rapidly to replace coal as a fuel for Naval purposes.

Since 1907 all torpedo-boat destroyers contracted for, of which there are twenty-nine, burn oil exclusively, and the battleships *Delaware, North Dakota, Florida, Utah, Wyoming, Arkansas, Texas* and *New York,* contracted for during this period, are fitted to burn oil as auxiliary to coal, each of these vessels carrying about 400 tons of the liquid fuel, to be burned at full power after the coal fires become dirty, or when it becomes difficult to trim coal from the bunkers into the fire-rooms. In the case of these battleships the advantages of the oil have so appealed to the *personnel* that oil alone is burned to

a great extent in port, and to some extent while cruising, although the installation of the oil-burning equipment did not contemplate these uses.

The *Nevada* and *Oklahoma*, the two battleships which have recently been contracted for, will burn oil exclusively. This is perhaps the most radical development in naval engineering since the advent of the turbine. It has permitted in the case of these vessels a reduction in boiler weights, which has made possible the use of heavier armor than has hitherto been employed. The reduction in length of boiler compartments has permitted the grouping of all boilers under one smoke pipe, which, of course, clears the upper deck considerably and permits more extensive arcs of fire for the turrets.

Aside from the use of oil as fuel under steam boilers, it now seems probable that within comparatively few years oil used in internal-combustion engines will furnish the principal fuel for all naval vessels. This is in consequence of the recent remarkable development of heavy-oil engines of the Diesel type in Europe. Hitherto oil engines have not merited much consideration for large naval vessels on account of the limited power that could be developed in a single cylinder. An installation of any considerable power required a multiplicity of cylinders. Now, however, we are credibly informed that 1,000 horsepower has been developed in a cylinder about 33 inches in diameter with a 40-inch stroke, at 150 r.p.m. in a 2-cycle marine-type readily-reversible engine. This engine has a speed control that is satisfactory, and an economy of fuel consumption probably twice that of a steam engine.

In the U. S. Navy heavy-oil engines built or so far projected are limited to a number of submarine vessels and to mother ships for submarines.

Gasoline is used as fuel for all of our earlier submarines and for a large number of small power boats carried by warships. Its use is likely to be discontinued entirely as soon as suitable heavy-oil motors for the small power boats are developed. As stated above heavy-oil engines are already supplanting gasoline engines in submarines.

# PETROLEUM AS A SOURCE OF POWER FOR SHIPS.

By Dr. Albert Sommer, Associate.

We are passing through a period of revolutionary events in the production of energy from fuel. With the steam engine solid coal is being burned under the boilers, water is evaporated, and the tension of the steam thus produced is transformed into mechanical motion. We may consider this method as the beginning of our age of mechanical propulsion, and while we see it still dominating the field, we are simultaneously witnessing the gradual introduction of the most developed stage, viz: the combustion of fuel in the cylinders of the machine itself. This latter principle, while not as yet perfected, must ultimately win the field, because its advantages and basic simplicity are too obvious to permit this development to receive a set-back. History shows that all obstacles in the way of necessary technical evolutions will be surmounted if their time has matured.

The burning of solid coal under boilers has been called criminal waste. Nothing could be more true than that, and the full extent of this squandering of natural wealth will be thoroughly realized only when the scarcity is felt by later generations.

Coal when taken from the mines loses in its subsequent stages of storing and handling an integral part of its energy owing to the escape of the more volatile constituents. It furthermore presents chemically a most complicated substance, from which matter valuable to human kind could be saved by using as fuel only such parts of it as are actually best fitted for that purpose. It has been suggested that all coal be coked at the mines, obtaining whatever by-products

of value the crude material might carry, and either make producer gas from the coke or use it locally for metallurgical purposes. The gaseous fuel could then be carried to its destination in pipes, and the other materials, simultaneously obtained, transported to their respective markets. Inasmuch as gas can today be carried under high pressure at a comparatively low cost, it might be surmised that this procedure will some day be adopted instead of carrying the coal by rail to its destination, where it is handled by much labor and its most valuable components burned regardless of their value—and mostly with considerable waste in economy.

This development, however, has received a temporary setback through the introduction of liquid fuel. While we are today witnessing attempts on a large scale by the great navies to introduce petroleum as fuel under their boilers, owing to the many and undeniable advantages connected with this innovation in military and economic regard; and while we see that the United States Navy, owing to the fortunate wealth of the American Continent in petroleum, is leading the world in this development; yet we must say from an engineering and scientific standpoint that this development must of necessity soon give way to the combustion of liquid fuel direct in the cylinders of the Diesel type of internal-combustion engines. We give below some actual data, recently obtained, illustrating the naval advantages obtainable in ships propelled by Diesel engines.

The Russian *Jakut* carried two Diesel engines of a total of 360 H.P., and also steam engines of the same power. The Diesel engines weighed 25 tons and the steam machinery 90 tons. This is a saving in weight in favor of the Diesel engines of 65 tons, or 70 per cent. A passenger boat of 207 tons displacement and carrying steam engines of 280 H.P., carried a load consisting of engine, boilers, etc., of 55 tons. The boilers and engines of this ship were replaced by Diesel engines of 310 H.P., the latter weighing 17 tons; so that 38 tons, or roughly 20 per cent. of the vessel's displacement, was saved by the introduction of Diesel engines. The same ship

had to carry a coal supply for one week, which, when in operation for 5 hours daily, amounted to 15 tons. Oil for the same period and extent of operation amounted to two tons. The saving in weight was therefore 13 tons, or 87 per cent. This does not, of course, include the saving which is accomplished by the elimination of the boiler water.

What is, however, very interesting from the standpoint of the Navy would be the advantages which could be gained if in the present ships all the deadweight now represented by engines, boilers, coal and water could be reduced by the introduction of Diesel engines, and the saving thus brought about given over entirely to the carrying of additional fuel. Calculations show that a ship could thus increase its radius of action eight times, or in all probability could make the trip round the world without being compelled to take on fresh fuel.

No matter what the development of the Diesel engine may be, and how soon its development may be accomplished—the ultimate forthcoming of which we do not entertain the slightest doubt—it is already a certainty that a ship equipped with Diesel engines will (*a*) be able to carry at least 30 per cent. more load; (*b*) will cost less to operate owing to the reduction in personnel, etc.; and (*c*) will be more economical to operate owing to the saving in fuel.

A Diesel engine for marine purposes works today with an efficiency of 35 per cent. compared with 8 to 10 per cent. of the steam engine. It can, therefore, be shown that with an oil of 19,000 B.t.u's. per pound compared with approximately 13,000 for coal, the price of oil may be five times as high as that of coal and it would be still more economical to operate a ship by oil engines than by coal. It is reasonable to conclude from these facts that when once the introduction of Diesel engines into the merchant marine, and probably more so in the Navy, has seriously begun, it will be an extremely difficult matter for the steam engine to compete with its younger rival; and it seems certain that a speedy adoption of the new principle, and even revolutionary changes, will be inevitable.

## AVAILABLE SUPPLY IN AMERICA.

We will now briefly discuss the available supplies of liquid fuel with a view of ascertaining to what extent, and how, they can be used.

The production of crude petroleum in the United States has gradually shifted from the east to the west, and has simultaneously shown a gradual change from oils of a pronounced paraffine character to asphaltic-base petroleum. California is today, with its asphaltic oil, supplying approximately one-quarter of the world's total production, viz: 250,000 barrels per day out of the world's daily total of 1,000,000 barrels.

To this will have to be added the production of Mexico, the future proportions of which are somewhat problematical, due to lack of accurate statistics; but there is every indication that this field, after the necessary arrangements have been made for thoroughly developing the production and the transportation of the oil, will assume equal importance to that of California.

Now, if we consider that liquid fuel in some form will represent in the near future the chief source of energy for naval propulsion, we are compelled to consider the asphaltic oils produced in the West and Central America as the chief materials from which the world will draw its supplies. At the present moment, if one looks over specifications which have been devised for liquid fuel in various countries, one cannot but be astonished at the restrictions which are imposed upon this commodity by the establishment of certain chemical limitations of doubtful value. The majority of restrictions are made with reference to the following factors: (*a*) flash point, (*b*) sulphur, (*c*) calorific value, (*d*) viscosity. Other specifications, which fortunately are not so generally adopted. include limitations to a certain gravity, the prohibition of mixed oils, or the limiting of the specified oil to a certain field.* If we consider that the world produces approximately 1,000,000,000 tons of coal per year, which is of the most

* Reference is made to a paper on "American Asphaltic Crude Oils and Their Significance in the Petroleum Industry of the Future," by the Author.

heterogeneous nature, ranging from lignite to anthracite, and that there has been no particular restriction in the use of this most variegated substance, it is almost mystifying to see restrictions established for a commodity like petroleum, which, being a hydro-carbon, can never, even in its most different forms, show such dissimilarity as often exists between two coals from neighboring mines. It seems to us, therefore, of vital importance to study the question whether, if the world, and especially the United States Navy, should consider the use of liquid fuel in some form necessary to its future development, it is justifiable to limit such material to a comparatively small field. Speaking commercially, it is obvious that such a policy would not be compatible with the adaptation of liquid fuel in its broadest scope, because in order to make petroleum the motive power for the ships of the world it must be available everywhere, as it will be, of course, necessary to use it everywhere; and if it should be compulsory either to manufacture this fuel by complicated processes or to restrict it to certain qualities only, one cannot see how its use could ever outlive the experimental stage.

### ASPHALTIC AND PARAFFINE OILS COMPARED.

The question will be asked, What is the essential difference between asphaltic and paraffine petroleums? It must be answered that there is no sharp difference. To the practical mind, which has, by the way, originated the terms paraffine base and asphaltic base, the most obvious difference is in the appearance, oils of paraffine base being comparatively thin and of a rather greenish and opaque appearance, whereas oils of asphaltic nature are black and viscous. The differentiation between the two is essentially American, inasmuch as oils produced in Russia and in the East Indies would fit in neither category. It would therefore be useless to attempt an exact classification of the two kinds of oils (which we migh rather describe as representatives of the extreme types of petroleum available in America), and content ourselves with the enumeration of some characteristic differences which these two oils

may possess. Several of the chemically pronounced features of each type are enumerated in the paper referred to in the preceding foot note as follows:

*Asphaltic Oils.*—(1) Their specific gravity is high for fractions between equal boiling limits; (2) their residues show a tendency to polymerize and solidify, especially in contact with catalytic agents; (3) they are generally high in sulphur; (4) they are subject to decomposition at a lower temperature than paraffine hydro-carbons; (5) they harden up under the influence of air at the higher temperatures; (6) their distillates have a comparatively high solvent power.

*Paraffine Oils.*—(1) They have a lighter specific gravity for fractions between equal boiling limits; (2) they are low in sulphur; (3) they are comparatively indifferent to air; (4) they do not polymerize at ordinary temperatures; (5) their distillates have a comparatively low solvent power; (6) they possess comparatively high capillarity.

The chief difference, therefore, would be apparent from the behavior of the two classes of oil against coarsely-distributed mineral fiber or filters, such as Fullers earth, charcoal, etc. The asphaltic substances are retained by such materials, whereas hydro-carbons belonging to the paraffine class would pass. This difference in behavior is largely if not entirely due to the fact that the substances in the paraffine oil are of crystalloid nature, whereas asphaltic bodies are colloidal substances.

Furthermore, in comparing asphaltic and paraffine oils, one should not believe that the presence of solid paraffine wax is the sole or even a very important characteristic of the latter kind. We give below the analysis of oils of decidedly paraffine character compared with some of asphaltic character, which will show that even oils of asphaltic character can contain solid paraffine wax, and, vice-versa, oils of paraffine nature can, and to a certain extent always will, carry a certain amount of asphaltic matter.

In the table are four American crude oils, representing an eastern oil, two mid-continental oils and a California oil. The eastern oil is a thoroughly representative paraffine-

| Test. | Pennsylvania crude. | Caddo crude. | Louisiana crude. | California crude. |
|---|---|---|---|---|
| Appearance | Greenish brown, liquid. | Dark brown, liquid. | Dark brown, liquid. | Black, viscous. |
| Specific gravity | 0.792 | 0.832 | 0.8955 | 0.9445 |
| Gravity Beaumé | 48° | 38.5° | 26.6° | 18.0° |
| Boiling point | 23° C. | 96° C. | 152° C. | 139° C. |
| Distillation (*non-cracking*): | | | | |
| At 100 degrees C | 6 per cent. | 1 per ct. | ... | ... |
| 150 degrees C | 23 per ct. | 6 per ct. | ... | 1 per ct. |
| 200 degrees C | ... | 21 per ct. | 8 per ct. | 4 per ct. |
| 250 degrees C | ... | 35 per ct. | 17 per ct. | 13 per ct. |
| 300 degrees C | 60 per ct.* | 48 per ct. | 41 per ct. | 22 per ct. |
| Paraffine scale: | | | | |
| (Holde method) | 2.8 per ct. | 2.5 per ct. | 0.35 p. ct. | 1.48 p. ct. |
| Asphalt: | | | | |
| By alcohol-ether | None. | 9.02 p. ct. | 2.85 p. ct. | ... |
| By petrol-ether | " | 0.04 p. ct. | 0.10 p. ct. | 14.98 p. ct. |
| Sulphur | 0.15 p. ct. | 0.30 p. ct. | 0.33 p. ct. | 3.3 per ct. |

**Cracking* distillation yields 20 per cent. more distillate at 300 degrees C.

base oil. The Caddo oil is typical of the mid-continental field, combining some features of both asphaltic and paraffine oils. The Louisiana oil is a peculiar substance, showing, as it were, very little of the characteristics of either class; its hydro-carbons, no doubt, are related to the asphaltic oils, but it carries very little asphalt and only a trace of paraffine. The California oil, however, is a typical asphaltic oil in all its features, but still carries a substantial percentage of paraffine; which is true of almost any asphaltic oil except some West Indian oils (Trinidad).

From the practical standpoint of an engineer who looks upon petroleum as a source of energy, these differences, notwithstanding their immense importance from the refiner's point of view and also from a chemical standpoint, are of secondary value. All he asks is, what energy is contained in such and such a fuel, and under what conditions can most of it be utilized for motive power. From this view-point, all petroleums are hydro-carbons, and, regardless of the chemical arrangement of the atoms in the petroleum molecule, he must

see that the total energy contained in or obtainable from petroleum is solely a question of the relative percentage of carbon and hydrogen contained in the same. The total energy due to the presence of these two combustible elements in petroleum can be measured by the calorific value of the oil; and this in turn has caused the establishment of specifications which prescribe a minimum for the calorific value as obtained in the bomb calorimeter.

DETERMINATION OF HEATING VALUES OF COAL AND OF OIL.

Reference is made to the success that has attended the use of heating-value specifications in buying and selling coal. This is a matter which has always been gravely doubted by men of practical experience. It is by no means an established fact that the combustion in the bomb yields results indicative of those obtained in actual combustion under the boiler, yet laboratory experts have developed a tendency to attribute an importance to the bomb determinations which they naturally cannot possess. The combustion in the bomb proceeds under high pressure, in an atmosphere of pure oxygen, and under conditions involving re-condensation of all water vapors produced by the combustion. The burning in the boiler not only proceeds under a draft instead of under a pressure, but also takes place in a multiple excess of oxygen, which in turn is accompanied by four times its volume of nitrogen; while the water vapors are not condensed but escape as gas. Consequently even a theoretical reflection will lead to the conclusion that the heat obtained in the bomb cannot parallel that which is produced under a boiler with the same kind of coal.

When we come to oil, however, this state of affairs is changed. Coal is a very complex aggregate of the most variegated hydro-carbons, combinations of nitrogen, of sulphur, and of inorganic matter; petroleum consists almost entirely of combinations of carbon and hydrogen, accompanied by very few other chemical elements, such as sulphur, traces of nitrogen, etc. With the exception of the sediment that the crude

oil carries and which can easily be removed, it contains no inorganic ash-forming components; hence the question of its value for practical burning purposes is very simply answered. No matter how the carbon and hydrogen may be combined, the practical heating value of such a combination will almost invariably depend on the relative percentages of carbon and hydrogen; hence all fuel oils, after the elimination of the water and sediment, appear as concentrated liquid fuel, all of which is combustible and the combustion of all of which proceeds along analogous lines. The relative percentages of carbon and hydrogen are, for all practical purposes, determined by the specific gravity of the oil, for, the atomic weight of carbon being 12 and that of hydrogen being 1, it is obvious that an oil containing the more carbon will be heavier than one in which there is less. It is therefore unnecessary even to undertake an elementary analysis of an oil so far as its practical value is concerned, for the simple determination of its specific gravity will, within narrow limits, indicate the relative percentages of carbon and hydrogen. This fact was established a long time ago by Kropff and Sherman, and, since their first publications, by other investigators, who have likewise established formulas for the calculation of the heating value of given oils from their gravity. From experience gained thus far it appears that these calculations tally with actual determinations of heating value in the bomb.

The formula established by Kropff and Sherman is as follows:

The heating value of an oil equals

$$18{,}650 + 40 \text{ (Gravity, Beaumé} - 10.)$$

For oils of pronounced asphaltic type such as California, formulas a little different have to be employed; but they vary only a little from one another, and it has almost invariably been observed that the results as calculated by these formulas tally within one per cent. with the results of a correct determination of the heating value in the bomb.

In this connection it should be stated that when determining the heating value in the bomb calorimeter, two products of combustion are formed, viz: $CO_2$ and $H_2O$. After the combustion carbonic acid is found as a gas under pressure, whilst the $H_2O$ vapor evolved during the combustion is found condensed as liquid water. Now, water possesses a latent heat of evaporation of 536 calories; in other words, in order to produce a pound of steam of 212 degrees F. from a pound of water of that temperature 536 calories are required; inasmuch as this addition of heat units cannot be detected by the thermometer, it is called the latent heat of evaporation. In recondensing the water vapors produced by the combustion this latent heat of evaporation for the quantity of water produced by the combustion is recovered, and we therefore obtain in the bomb a higher heating value than if the water were retained in its gaseous form in quantities commensurate with the percentage of hydrogen present in the original oil. The result obtained in the bomb when burning a hydro-carbon is therefore called gross heating value, in contrast to a heating value which would be obtained if the water were retained as vapor, which is called the net heating value. In the case of the combustion of a gaseous hydro-carbon, the difference between the two values as established in the calorimeter is as much as 10 per cent. or even more; whereas with a liquid such as petroleum it is about 7 per cent.

In actual combustion, so far as the power producing unit is concerned, water escapes invariably as a gas; therefore the value as obtained in the bomb, or even as calculated by the formula, does not tally exactly with the true value of an oil, pound per pound. Taking the *gross* heating value as a criterion for the energy contained in an oil, it is of course true that the lighter fuel will contain more energy; but it must be noted that with the heavier oil, owing to its lesser proportion of hydrogen, the *net* heating value is nearer to the theoretical or gross heating value. When oil is purchased by *volume,* the actual differences in consumption for equal power produced by light oils and heavy oils are very small.

Below we give a table showing some exact determinations of the heating value of oils of various origin. These figures were obtained from the Mahler bomb. The water condensed was determined by chemical analysis, so that the net heating values are derived analytically and not by calculation.

| | Roumanian gas oil, No. 1. | Roumanian gas oil, No. 2. | British Navy fuel. | Roumanian oil. | Trinidad liquid asphalt. | Tarakan crude. | Roumanian residuum. |
|---|---|---|---|---|---|---|---|
| Specific gravity | 0.868 | 0.871 | 0.907 | 0.928 | 0.964 | 0.948 | 0.930 |
| Beaumé | 31. | 31. | 24. | 21. | 15. | 18. | 20.5 |
| Ash, per cent | ... | ... | 0.023 | 0.051 | 0.14 | 0.035 | 0.13 |
| Water, per cent | Trace. | Trace. | 0.5 | Trace. | Trace. | Trace. | Trace. |
| Gross heating value per pound: | | | | | | | |
| Calories | 10,697 | 10,712 | 10,695 | 10,510 | 10,224 | 10,487 | 10,558 |
| B.t.u's | 19,255 | 19,282 | 19,252 | 18,918 | 18,405 | 18,873 | 19,008 |
| Net heating value per pound: | | | | | | | |
| Calories | 9,971 | 10,003 | 9,943 | 9,823 | 9,574 | 9,804 | 9,848 |
| B.t.u's | 17,946 | 18,009 | 17,901 | 17,684 | 17,235 | 17,649 | 17,730 |
| Difference, per cent | 6.79 | 6.62 | 7.03 | 6.54 | 6.36 | 6.51 | 6.72 |
| Sulphur, per cent | ... | ... | ... | 0.43 | 3.36 | 0.25 | ... |
| Gross heating value per gallon: | | | | | | | |
| Calories | 77,339 | 77,662 | 80,747 | 81,200 | 82,062 | 82,742 | 81,720 |
| B.t.u's | 139,211 | 139,791 | 145,345 | 146,160 | 147,711 | 148,936 | 147,104 |
| Net heating value per gallon: | | | | | | | |
| Calories | 72,090 | 72,522 | 75,069 | 71,315 | 76,841 | 77,353 | 76,223 |
| B.t.u's | 129,762 | 130,539 | 135,125 | 138,367 | 138,314 | 139,236 | 137,202 |

Inasmuch as in Diesel engines the efficiency of each oil is in direct accordance with its heating value, the greatest theoretical deviations in the efficiencies of various oils can be only in direct proportion to their heating values; therefore the following table will be of interest, not only from the standpoint of the relative heating values *per se*, but also from the standpoint of their relative efficiencies in internal-combustion engines. In this table eight different petroleums. ranging in gravity from 0.871 to 0.964, or from 31.0 degrees to 15.5 degrees Beaumé, have been actually tested for heating values, and furthermore the same have been calculated. The

heating value of the lightest oil, viz: the Roumanian gas oil of 0.871 specific gravity, has been assumed to represent 100 per cent., and the others have been calculated in relation to this. It will be seen from these figures that the heaviest of them all, viz: the Trinidad residuum, of specific gravity 0.964, or 15.5 degrees Beaumé, by actual experiment represents 95.4 per cent. of the Roumanian gas oil in efficiency, and theoretically, by the formula, 96.4 per cent.; so the greatest deviation possible between an oil of 0.871 specific gravity and one of 0.964 specific gravity amounts to less than 5 per cent.

| ..... | Specific gravity. | Beaumé. | Gross heating value by experiment. | | Per cent. | Gross heating value calculated by formula. | | Per cent. |
|---|---|---|---|---|---|---|---|---|
| | | | Calories. | B.t.u's. | | Calories. | B.t.u's. | |
| Roumanian gas oil. | 0.871 | 31.0 | 10,712 | 19,282 | 100 | 10,827 | 19,490 | 100 |
| Admiralty fuel. | 0.907 | 24.25 | 10,696 | 19,253 | 99.8 | 10,677 | 19,220 | 98.6 |
| Roumanian fuel. | 0.927 | 20.95 | 10,557 | 19,003 | 98.5 | 10,604 | 19,068 | 97.9 |
| Roumanian residuum. | 0.928 | 20.8 | 10,558 | 19,004 | 98.5 | 10,601 | 19,082 | 97.9 |
| Trininad crude oil. | 0.945 | 18.05 | 10,200 | 18,360 | 95.2* | 10,540 | 18,972 | 97.3 |
| Roumanian residuum. | 0.946 | 17.9 | 10,510 | 18,918 | 98.1 | 10,536 | 18,966 | 97.3 |
| Tarakan crude oil. | 0.948 | 17.6 | 10,487 | 18,877 | 97.8 | 10,630 | 18,954 | 97.2 |
| Trinidad residuum. | 0.964 | 15.5 | 10,224 | 18,405 | 95.4* | 10,447 | 18,606 | 96.4 |

The dropping off of the Trinidad residuum and Trinidad crude from the theoretical value is doubtlessly due to the high percentage of sulphur—sulphur having a low heating value. These two oils are marked with an asterisk.

All the heating values given above are for weight. In purchasing by volume the heating value of a heavy oil would be higher than that of a lighter oil, because the weight in pounds per gallon more than offsets the decrease of heating value per pound, so that the same oils as enumerated in the foregoing table would show heating values per gallon as follows:

| Oil. | Gross heating value by experiment per pound, B.t.u's. | Gross heating value per gallon, B.t.u's. |
|---|---|---|
| Roumanian gas oil | 19,282 | 139,794 |
| Admiralty fuel | 19,253 | 144,975 |
| Roumanian fuel | 19,003 | 146,693 |
| Roumanian residuum | 19,004 | 146,901 |
| Trinidad crude oil | 18,360 | 144,499 |
| Roumanian residuum | 18,918 | 149,073 |
| Tarakan crude oil | 18,877 | 148,940 |
| Trinidad residuum | 18,405 | 147,793 |

When burning oil instead of coal one factor has not generally been sufficiently appreciated; that is the fact that with oil the actual temperature of the flame, or the pyrometric effect, is very much higher than that attained with coal. This is due, of course, to the fact that oil represents a much more concentrated fuel than coal; that it contains a large percentage of active hydrogen; and that it is not necessary to heat an appreciable excess of air—which always, of course, carries four-fifths of inactive nitrogen. The temperature of a flame, or its pyrometric effect, is directly proportionate to the heating value of the fuel and inversely proportionate to the quantity of gaseous products of combustion or waste gases and the specific heat of such gases, while the decomposition of the products of the combustion has to be considered.

Owing to the high temperature which can be reached in an oil flame, the use of liquid fuel is of special advantage, and can probably bring about some improvements, which we do not know as yet, in metallurgical processes and in all such places where a high potential of heat is required.

Again, especially so far as use on board ship is concerned, it must be remembered that with coal most of the combustion takes place on the grate itself with air being continuously sucked through the bed of coal, whereas with oil combustion takes place solely in the fine particles of oil which issue from the fuel nozzle; and that, owing to the great richness of this gas compared with that issuing from coal, a comparatively greater amount of air is necessary with the flame itself. For

instance, while a ton of bituminous coal would yield about 10,000 cubic feet of gas, a ton of oil would yield approximately 20,000 cubic feet of gas, which latter, being far richer in hydro-carbons than that issuing from the coal, requires a far greater volume of air for its complete combustion. It can easily be understood, then, that without extensive alterations a furnace with its combustion area and air draft arranged for coal cannot be expected to deal with nearly the same weight of liquid fuel.

This, of course, has nothing to do with the above-mentioned *excess* of air required in a coal fire.

## SULPHUR.

The sulphur in petroleum has been a regular bugbear ever since the introduction of crude petroleum or its various constituents into the production of power. The British Navy has established a maximum of $\frac{3}{4}$ per cent. of sulphur in the oil, and so has the United States Navy. There is no doubt that no scientific reasons can be given for setting this or any other limit to the sulphur contents of an oil. The fears which are entertained in regard to the damaging influence of this element are the following: (*a*) corrosive influence of the flue gases on the boilers; (*b*) corrosive influences on the flues; (*c*) causing of deposits together with lamp-black of corrosive influence; (*d*) finally, and probably most important, corrosive effects of the gases of combustion when the oils are used in internal-combustion engines.

The importance of this issue just at the present moment is very great. The daily production of California oils is, as stated above, 250,000 barrels; and there are 45,000,000 barrels in storage awaiting transportation, practically all of which is of asphaltic character. We are also on the eve of great developments in Mexico, the oils of which are even higher in sulphur than California oil, often showing 3 and even more per cent.

It would be impossible to consider the use of fuel oil in the

near future on a large scale if we were forced to eliminate these oils.

In asking ourselves whether there is any justification for limiting the percentage of sulphur, we must state beforehand that absolutely no reason can be given for fixing a figure like 1 per cent. or $\frac{3}{4}$ per cent., because nobody can tell why an oil containing less sulphur than that should be harmless and one containing more should be damaging. A canvass extending over the past two years has failed to bring to light any practical facts elucidating why such a limit should be set other than by pure guess-work.

Owing to the great preponderance of asphaltic production today, and to the coming large supply of liquid fuel to the world when the transportation facilities are enlarged by the commissioning of the many tank steamers now on the stocks, it is necessary that each consumer of this commodity should satisfy himself as to the justification for each limitation he puts on his fuel in establishing specifications, and this especially with regard to the sulphur clauses.

During the ordinary combustion of oils the sulphur turns completely into sulphurous acid, provided that there is sufficient excess of air. If the flame has insufficient oxygen, it is a well-known fact that sulphuretted hydrogen and sulphur might be produced in the gases, and these substances when coming into contact with incandescent iron combine to iron sulphate. This reaction, of course, would soon cause the iron to completely lose its stability. It is, however, a very rare case for a flame to be reducing to such an extent that such gases can be formed. There are no experiences to show at what temperature such a reaction would begin to any damaging extent; but it being an easy matter to regulate the flame so that there will always be a sufficient supply of air to produce *complete combustion,* we may take it for granted that the flue gases due to the presence of sulphur in the oil will consist only of $SO_2$ and $SO_3$.

It is a proven fact that sulphurous acid ($SO_2$) and sulphur trioxide ($SO_3$) do not attack iron unless liquid water is pres-

ent. Experience has shown that the products of combustion resulting from burning oil which contains up to ¾ per cent. of sulphur do not attack steam boilers to a noticeable extent. It can therefore be concluded that sulphuric and sulphurous acid gas and water vapor do not act on iron at the temperatures which exist in steaming boilers. Theoretically, there is no reason why a percentage of sulphur in excess of ¾ per cent. should render an oil dangerous for boilers. Rather should it be expected that a mixture of gas containing much more sulphurous acid would, at the temperature of a boiler in operation, act exactly as would flue gas produced from an oil containing ¾ per cent. or less of sulphur.

Provided, therefore, that the flue gases always carry an excess of oxygen, there is no reason why sulphur should act in any way differently when more than ¾ per cent. is present in the oil.

A source of danger, however, would lie in the possibility of condensation of the water contained in the flue gases. In ordinary combustion and under ordinary conditions the flue gases would leave the funnels at a temperature far above 212 degrees F., so that there could be no danger of condensation. But there might be corners in which lamp black or other solid substances would be thrown down, which, owing to their porous character and large surface per volume, would act as catalytic agents, causing surface condensation of water with the sulphurous acid; and this would exercise a direct corrosive influence on the spots where it accumulated. It has been found that lamp-black thrown down in inaccessible places in the flue channels is always accompanied by high amounts of sulphuric acid which has formed by gradual accumulation. It is therefore necessary to have the flue channels free from sudden bends, traps, or other inaccessible places where such deposits can form.

It is, however, obvious that the same corrosive action could occur when the sulphur in the oil is less than ¾ per cent., inasmuch as the accumulation of acid deposits would be a gradual and not a sudden one. It follows that under all con-

ditions attempts should be made to completely burn up the soot; in the case of oil fires this is a very easy matter. The combustion of soot is possible if, in the case of large flues, their front parts are coated with fire brick; or in the case of a great number of smaller tubes the furnace is so constructed that the main combustion takes place in a separate fire box lined with fire brick.

No objection has ever been taken to the sulphur contained in coal, although coal containing large amounts of sulphur is burned all over the world; and there has never been an observation of the increased deterioration of the boilers on that score. If one considers that the heating value of the average oil is almost twice as high as that of coal, pound for pound, and calculates the amount of sulphur contained in an ordinary coal on a basis of equal heating values of coal and oil, one arrives immediately in coal at figures of 5 or 6 per cent., or even more, for the corresponding B.t.u's. of oil, which amount in the case of oil would cause great anxiety.

One objection to the use of coal containing sulphur is its bad effect on the grates. The sulphur in the coal combines with the iron and gradually impairs its stability. This source of trouble is, of course, completely eliminated with oil.

A coal containing, for instance, 5 per cent. of sulphur, as many of them do (some even up to 10 per cent.), would form sulphurous or sulphuric acid amounting to 15 per cent. or more of the weight of the coal, which would, of course, eat up the steam boiler fired with such coal in the shortest possible time. The only reason that this does not occur lies in the fact that the flue gases leave the flues with a temperature of from 400 to 600 degrees F., at which temperatures no sulphuric acid can be condensed. This, of course, holds equally good for oil.

The only influence that sulphur can have on oil is due to the fact that it has not quite the heating value of carbon, and therefore a large percentage of sulphur in oil will naturally depress its heating value. The depression is, with the percentage of sulphur in average asphaltic oil, even in Mexican

and California oils, so little that for practical purposes it need hardly be considered. Mexican, as well as California factories, ships and railways, are using enormous quantities of high-sulphur oils today with the greatest economy; and in a very careful canvass which the writer has had taken in California and also in Mexico he has not been able to establish one case of damage or complaint due to the actual influence of sulphur.

We are giving below a table showing results of a personal inspection of several large plants where California asphaltic oil is burned, and of the subsequent analyses of samples, all

| Consumption, in barrels, per year. | Burner used. | Sample from— | Sulphur in sample, per cent. |
|---|---|---|---|
| 12,000,000 | Pat Sheedy burner, made by J. C. Martin & Co., Bilboa Building, San Francisco. Gravity feed on locomotives with 1 foot fall. Stationary boilers, 40 pounds pressure, with admission of air to increase combustion. | Storage tank. | 0.73 |
| About 675,000 | Not stated; steam pressure, 150 pounds. Oil, 50 lbs. | Steamer ..... | 0.95 |
| 500,000 | Hammel burner at 25 to 40 pounds pressure. Ratio of steam pressure to oil pressure, 3 or 4 to 1. Ratio sustained by regulator. | Oil-feed pump. | 2.81 |
| 500,000 | Thomas. Van Bourtan & Ingersoll. (Are trying out; the "Thomas" favored.) | Engine tank.. | 1.03 |
| 9,600 | Hammel at 33 pounds pressure with steam at 150. | Oil-feed pump, | 2.16 |
| 4,800 | Booth burner (warm oil flows on to steam jet). They bought the patent and make it themselves. Gravity feed with 1 foot fall as minimum. | Engine tank.. | 2.03 |
| Not stated. | Hammel locomotive burner; gravity feed. | Engine tank.. | 2.18 |

of which were taken directly from the places of use. For obvious reasons no names of these places can be quoted; but the advantages of the oil throughout were highly praised.

No doubt in future a more important side of the sulphur issue will consist in the question of its danger to the cylinders of internal-combustion engines. Some of the data given below in this connection is quoted from a recent article on the subject by Dr. Edmund Graefe in No. 2 of the "Oelmotor" (May, 1912), entitled, "Ueber den Einfluss des Schwefels in flüssigen Brennstoffen beim Motorenbetrieb."

In the cylinders of internal-combustion engines, where the most vital parts, combustion valves and cylinder walls, are in constant direct touch with the products of combustion, a corrosion of such surfaces would, of course, be of vastly graver importance than an attack on parts of a steam boiler. In the case of gas engines, great fears were at first entertained in regard to the sulphur in gas; but nothing was heard of damage to gas motors from this source.

With the introduction of oil motors the extent of the sulphur in the liquid fuel was in the foreground of interest. Today great anxiety is felt by prospective users of Diesel engines in regard to the economy of their supplies, which, of course, would be greatly impaired if the oils carrying a high percentage of sulphur were barred. Probably the latest experience in regard to sulphur oils used in internal-combustion engines—or, as we will call them here, Diesel engines—has been available in Germany, and we are quoting below Dr. Graefe's statements in this connection.

In Germany there exists a high import duty on all petroleum products, and therefore what are generally in use are distillates obtained from brown coal tar, or shale, available in mid-Germany, the tar of which yields products of distillation greatly resembling petroleum products. These distillates contain approximately 1.5 per cent. or more of sulphur. Considering that they have a calorific value of approximately 10,500 calories (19,000 B.t.u's.), as compared with 5,000 of ordinary illuminating gas, their percentage of sulphur com-

pared with the gas when figured for an equal heating value would be ten times as great. These oils were used in Germany, before the introduction of the Diesel engines, for the operation of power generators based on the explosive principle. When the Diesel engines were introduced the only economical oils available were the heavy oils produced from the brown coal tar, with a specific gravity of 0.860 to 0.920 (33 to 22 Beaumé), which carry approximately the same heating value as the above-named solar oils, and contain up to 2 per cent. of sulphur. The use of such oils started in the years 1902–1903, and when any irregularities occurred to these machines the sulphur was invariably blamed as the cause; but in no case could this be substantiated, the damage always being successfully traced to other causes.

The products of combustion of sulphur, $SO_2$ and $SO_3$, do not pre-exist in the oil. It is not known exactly in what combination sulphur is contained; but it has never been observed that it was contained in the oil in such combination as had the character of an acid or could contribute to the corrosion of metal. All of the combinations, rather, in which sulphur exists are inert and entirely harmless to metal. In Diesel engines the sulphur burns chiefly as follows:

$$S + O_2 = SO_2,$$

but a part of it, especially on account of the high pressure produced in the motor, burns as follows:

$$2S + 3O_2 = 2SO_3,$$

and this yields sulphuric acid anhydride. These two products are by themselves absolutely harmless to metal, as is amply shown by the fact that they are transported in their dry condition in iron receptacles. Immediately, of course, they have an opportunity to dissolve in water they form sulphurous acid and sulphuric acid, both of which are highly corrosive to iron. To this end, however, it is necessary that they become condensed as a liquid, and this requires a sufficient cooling below the boiling point which never takes place in the interior of a Diesel engine during its operation.

The impact on metal caused by sulphurous products of combustion is therefore conditional upon the condensation of water vapor, which absorbs the acid oxides of sulphur. Water originates through the combustion of hydrogen, which is an integral component of the petroleum. Those oils which are generally used contain approximately 11 to 13 per cent. of hydrogen, which yields 9 times its amount of water during the combustion; so that a pound of oil produces only 1.2 pounds of water. During normal operation of a motor the condensation of water in the cylinder is out of the question. There have been various other most interesting causes of trouble in connection with sulphur which Dr. Graefe has personally observed and reports on as follows:

It should be stated in this connection that these motors were installed in various mines of the greatest German brown coal concern, and all of them were operated with brown coal tar oil.

The motors began to show trouble in operation, working slower and slower, and finally stopping altogether. They were taken apart and, to the surprise of the mechanics, no trace of trouble inside the cylinders could be found. A diagram was subsequently taken on the motors, which showed that the exhaust pressure was extremely high, which gave cause to thoroughly investigate the exhaust conduits. This revealed that in all cases the exhaust tubes were choked, most of them, with masses as hard as stone. Chemical examination of this substance showed that it consisted largely of basic ferric-sulphate, and also of ferrous-sulphate. Below is a table embodying the results of the analyses of such deposits in the exhaust pipes in 14 cases.

Nearly all the motors which showed this trouble had either long exhaust conduits or such as were exposed, and the trouble occurred nearly only in winter time. Owing to the cooling around the length of the exhaust conduits, the temperature of the exhaust gases was sufficiently reduced for water to begin to condense, which in turn absorbed the products of combustion, $SO_2$ and $SO_3$, which ran along the tube as

| Number. | Appearance. | Soluble in water. | Insoluble. | Ashes of the insoluble (calculated for insoluble matter). | Soluble $FeSO_4$. | Total results. | |
|---|---|---|---|---|---|---|---|
| | | | | | | $Fe_2O_3$. | Volatile matter (brown oil and $SO_3$ from the basic ferric sulphate). |
| | | *Per cent.* | *Per cent.* | *Per cent.* | *Per cent.* | *Per cent.* | *Per cent.* |
| 1 | black..... | 88.33 | 11.67 | 36.4 $Fe_2O_3$ | 77.5 | 4.25 | 7.42 |
| 2 | light gray. | 55.16 | 44.84 | 65.7 | 32.0 | 29.47 | 15.37 |
| 3 | dark gray. | 81.75 | 18.25 | 38.7 | 52.1 | 7.07 | 11.18 |
| 4 | gray ..... | 64.73 | 35.27 | 57.3 | 74.3 | 20.02 | 15.07 |
| 5 | dark gray. | 26.98 | 73.02 | 63.5 | 21.0 | 46.04 | 26.62 |
| 6 | gray ..... | 50.38 | 49.62 | 69.5 | 41.2 | 34.05 | 15.09 |
| 7 | black ..... | 82.31 | 17.69 | 40.8 | 28.5 | 7.22 | 10.47 |
| 8 | black, brown, gray ... | 32.14 | 67.86 | 75.5 | 75.8 | 5.12 | 16.07 |
| 9 | slate gray. | 70.65 | 29.35 | 68.0 | 49.4 | 19.09 | 9.42 |
| 10 | black..... | 80.65 | 19.35 | 42.2 | 77.7 | 8.15 | 11.20 |
| 11 | gray ..... | 85.25 | 14.75 | 35.2 | 78.5 | 5.19 | 9.56 |
| 12 | black ..... | 4.29 | 95.71 | 82.2 | 2.7 | 78.57 | 17.14 |
| 13 | black, gray .... | 89.01 | 10.99 | 37.6 | 82.5 | 4.11 | 6.88 |
| 14 | gray ..... | 74.54 | 25.46 | 62.0 | 57.0 | 15.75 | 9.71 |

dilute sulphurous and sulphuric acid. These solutions, of course, attacked the iron, forming ferro-sulphate; the sulphurous acid also of course, on contact with the air, quickly oxidized to sulphuric acid. Then the solution of ferrous-sulphate ran along the tubes, meeting hot exhaust gases, the water being gradually evaporated and the solution concentrated. Thus gradually the iron vitriol began to crystallize, and by-and-by caused complete clogging of the tubes.

Immediately after the removal of such deposits the motors worked again without any difficulty. It is shown that such corrosion which may occur on account of sulphur is easily avoidable if one endeavors to remove the causes. In one case the exhaust pipe was reduced in length from 30 feet to 6 feet, whereupon there was no further trouble in connection with the operation of this engine, because the exhaust gases left the tube hot enough to experience no condensation. Where, however, it is not possible to shorten the exhaust tube, a lead tube may be inserted in such conduits; lead will not be attacked by the dilute acid. In this case, of course, care has to be taken

that the condensed water is drained, otherwise it would attack the dashpot. It is also practicable to use clay pipes within the iron exhaust tubes.

Another source of trouble can occur with motors using a part of the compressed air in the cylinder for the oil spray. This system is now antiquated, and such cases are consequently principally of an historic interest, but nevertheless deserve to be noted. In these motors the compact air which is produced by the compression stroke of the cylinder is led into separate receptacles where it is compressed further and then used for forcing the oil into the combustion cylinder. Such air as is drawn from the compression stroke of the cylinder of course always contains a certain percentage of waste gases, which tend to accumulate gradually in the receptacles used for storing such air. Their condensation will take place owing to the cooling and especially owing to the high pressure, and thus sulphurous and sulphuric acid will form there, which will cause corrosion of the receptacles. This can also, of course, be combated by lead-coating the inside of the receptacle.

It is clearly seen by the results of this investigation that sulphur evidently is no disturbing factor where it has been dreaded most, viz: in the cylinders and valves of internal-combustion engines, but that naturally due consideration has to be given to the presence of sulphur in such oils in order to protect the exhaust conduits and all such parts in which condensation might take place. In Germany most of the Diesel engines are now operated with tar oils produced from ordinary coal, which contain 2 per cent. and often much more of sulphur; nevertheless these are used on a very large scale.

The importance of these products for the internal-combustion engines, owing to the prohibitive duty on petroleum products imported into Germany, has assumed such dimensions that special types of motors have been devised which successfully operate with tar products; but no case of damage due to sulphur has been observed.

In the meantime, extensive experiments have been made

under the supervision of the author in using high-sulphur oils for Diesel motors, in the course of which a Trinidad asphaltic residuum which carried over 3 per cent. of sulphur was used several days in succession. The combustion of this substance, aside from its unusual consistency (it being semi-solid), to which we will refer further below, proceeded without the slightest difficulty. The exhaust gases were absolutely clear, and the engine after being taken apart showed not the slightest traces of corrosion or other damaging effects due to the sulphur. On the contrary, it could be distinctly noticed that, owing to the comparative readiness with which these asphaltic substances break up, the combustion took place with ease, and there was no tendency to form deposits or other causes of trouble.

## CONSISTENCY OF OILS.

The remainder of this paper will be devoted entirely to the question of the consistency of fuel so far as its use in internal-combustion engines is concerned.

It is obvious that in ordinary burners any hydro-carbon can be used provided that it can be melted and brought to sufficient liquidity to be pumped through the burner. For Diesel engines, however, great apprehension has been and still is extant regarding the use of the heavier residuums, which could not, according to universal belief, be burned successfully in Diesel engines. Some engineers disclaimed any usefulness for the heavier residuums altogether, while others admitted that they could perhaps be used in stationary engines which could be cleaned at brief and regular intervals; but that, owing to the fact that heavier residuums would cause corrosion and deposits in the cylinders, valves and atomizing instruments, they could not be used on ships where no stopping of the engines was feasible, the deposits gradually accumulating under such conditions until the engine stopped altogether. Authorities on these subjects are on record, in publications, who have crude petroleum divided into several portions, the heavier of which were thrown out as useless for internal-combustion engines.

In this connection it seems to have been totally forgotten that the first Diesel engine constructed was not operated with liquid fuel, but with coal dust; and that the only reason why the use of this fuel did not continue was the fact that all coal, and even charcoal, contains a certain amount of ashes, which, of course, after the combustion of the organic part, would abrade and ultimately destroy the cylinders, pistons and valves. In the case of any petroleum, however, no matter what its consistency, and even with solid pitches originating from petroleum, no ashes to any extent can be formed; and consequently, theoretically speaking, it must be possible to burn petroleum of any description in an internal-combustion engine without any trouble whatsoever.

Nevertheless, in practical use deposits have been formed. In some cases the valves have been gradually covered with a black pitch which ultimately became so thick that it prevented further operation of the machine; or the cylinder walls were coated with a varnish-like deposit; or the piston rings became encrusted so that there was no more spring or slide in them, gradually decreasing the efficiency of the engine until the piston was so slack that the power of the explosion ceased. In some cases it is reported that " sausages" of solid carbon were formed, which gradually issued from the cylinders, clogging the same up entirely.

It was stated that all kinds of expedients had been used to overcome this feature, but without avail. The compression air was increased; the pressure of the secondary air used for the injection of the oil was increased up to 90 atmospheres; in two-cycle engines the original compression was increased by checking the exhaust; all without success.

In a recent investigation which has taken place under the supervision of Dr. Edmund Graefe and the writer,* these conditions have been looked into thoroughly and followed out to the minutest detail; and while the results of this investigation will be published *in extenso* elsewhere, some of them may be briefly quoted here.

The first question in commencing an investigation into this

* This investigation was carried out for Mr. F. Lane, of London.

subject was, of course, What circumstances can cause these deposits? They might be the consequence of some gradual evaporation, or distillation or "cracking" of the oil after it has reached the cylinder, in such a manner that lighter hydrocarbons will burn off and a remainder will form in which the carbonaceous character is more expressed, and which consequently will represent a hydro-carbon, the percentage of carbon in which is greatly increased or predominant, or which even represents free carbon or coke. In other words, particles of the oil entering the cylinder must undergo a similar process to an oil in an oil-gas retort where highly carbonaceous products are formed in the shape of oil-gas tar. It is, of course, an obvious thought that these products might be formed by the oil striking the cylinder walls or valves, and that distillation might there and then take place. It is, of course, not impossible, especially in case the cylinder walls are cooled too vigorously by the cooling water, that such a reaction might take place on them; but in ordinary combustion in a Diesel engine cylinder by the time the oil has travelled that distance it should be completely consumed.

Another possibility, however, would consist in the particles of oil going through the atomizer being comparatively large and viscous, in which case will take place all over their surface a vigorous and simultaneous combustion and forming of a carbonaceous residue, which, by the time the surface of the particle is consumed would, owing to the decrease in pressure which has taken place in the meantime, remain non-combustible; and on being thrown against the metallic parts of the cylinder they would gradually accumulate and form the deposits found in practice. In other words, if the particles of the oil entering the cylinder could always be selected so that their volume would be a minimum, so that the time required for their complete combustion is not great enough to permit of the formation of a non-combustible residuum, all trouble caused by deposits of any heavy petroleum residuum would be avoided.

It is this which has been proved beyond further doubt by

the experiments above referred to. Chiefly four oils have been used for the purpose, the viscosities of which at different temperatures are shown in the following table:

VISCOSITY AS DETERMINED BY THE ENGLER APPARATUS.

| Oil. | Specific gravity. | Seconds taken by 200 c.cm. to flow out at a temperature of C. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 20° | 30° | 40° | 50° | 60° | 70° | 80° | 90° |
| British Admiralty fuel..... | 0.907 | 818.5 | **435.3** | 255.0 | 178.9 | 139.9 | 109.5 | ... | ... |
| Roumanian residuum ..... | 0.946 | ... | ... | ... | 784.7 | **473.7** | 292.5 | 199.0 | ... |
| Tarakan crude oil ..... | 0.948 | 745.5 | **431.0** | 263.1 | 169.7 | 127.0 | 99.2 | ... | ... |
| Trinidad asphalt residuum | 0.964 | ... | ... | ... | ... | ... | 1,126.8 | 637.5 | **445.2** |

A glance at the viscosities printed in heavy type will show the great difference in the temperatures necessary to bring these four oils to approximately the same viscosity.

The more viscous of these oils could not at first be burned without trouble in the cylinders of the engines which served for the experiments; but after giving them the necessary degree of viscosity which was required in accordance with their individual characters—which was attained by means of heating them by an appropriate device—not the slightest further trouble was experienced with any of the above-named residua or any other, no matter how viscous it was. With some of the above oils the same motor has been running day and night for 18 days without the slightest trouble either from deposit or corrosion; and should the motors have been available for a longer period it was quite apparent from the condition of the valves, atomizer, piston and cylinder walls, after the engine had been taken apart at the end of that time, that it could certainly have run for the same period again without cleaning becoming necessary. No priming oil was used and no complication of the operation was introduced, which would, of course, have seriously questioned the value of any innovation in ordinary marine use.

It can thus be stated that the consistency of any petroleum,

no matter what it might be, provided it has been freed from mechanical impurities, will not be of any moment so far as its use in internal-combustion engines is concerned; and that the world will no doubt see the use of the heaviest asphaltic residua with comparatively high amounts of sulphur in internal-combustion engines, which will render their use of course immediately more popular and general.

# THE MANUFACTURE AND USE OF STEEL TUBING.

By J. H. Nicholson, Associate, and Emil Holinger.*

Before taking up our real subject matter I wish to make a few general remarks concerning the pipe and tube industry that may be of interest. In the United States the amount of pipe and tubes produced annually is about 2,500,000 tons. Of this amount only about 250,000 tons are used for boiler manufacture; the remainder for a great variety of purposes. Almost one-half of this immense tonnage is produced by the National Tube Company, with which company we are both connected. They employ about 17,000 men, have modern, up-to-date, self-contained plants, and control all parts of the operation of manufacture from the ore to the finished product. The tube industry has had a steady and remarkable growth during the last twelve years, the production having increased from not more than 750,000 tons to 2,500,000 tons during that time. The demands for pipe in developing our natural resources of oil and gas in this country have had great influence on this growth, as their production and transportation is of great economic importance. Inventive ingenuity and skill, with modern methods of manufacture and operations, have resulted in such cheapening of costs of production, that today there is no merchantable article of iron or steel sold at such proportionately low prices as pipe, when labor costs and complexity of fabrication are taken into consideration.

## PROCESSES OF MANUFACTURE.

In taking up the first section of this talk, the manufacture of pipe and tubes, I am only going to briefly describe the

* Extracts from an address to the U. S. Naval School of Marine Engineering, Annapolis, Md.

process of manufacture, as technical books, trade papers and advertising matter have made such information, with illustrations and well-written descriptions of process, equipment and operations so generally available that you are all doubtless familiar in a general way with the subject, and some of you have probably visited different works and seen the actual operations of manufacture. It will be the endeavor of myself as well as my associate who will take part with me in this talk to utilize most of the available time on some portions of the subject not generally dealt with in technical books or trade papers, hoping that some of it at least will be new and of engineering interest and value.

The manufacture of tubes can be divided into two general classifications—welded and seamless. Welded tubes can be divided into two sub-classes—butt and lap. Both are made from strips of rolled iron or steel, commercially called skelp. Butt-welded are made in sizes from $\frac{1}{4}$ inch to $3\frac{1}{2}$ inches in diameter; lap-welded from 2 inches to 30 inches in diameter. The difference between these two kinds of tubes is in the method of making. In the butt-welded tube the square edges of the skelp when at a welding heat are brought together by pressure, and welded when the strip is drawn through a circular die, without the use of an interior support or mandrel. In the case of the lap-welded tube the edges of the skelp are scarfed and bent in tubular form so as to lap one edge over the other, and, when brought to a welding heat, is passed through a circular groove in the welding rolls with a supporting cast-iron ball or mandrel on the inside, producing pressure upon the overlapped scarfed edges, welding them and reducing the thickness of the lap to the uniform gage of the rolled skelp.

Referring now to the description of the manufacture of seamless tubes, we can divide their manufacture into four different processes:

First, the punching of solid billets by hydraulic or power presses, and the subsequent rolling or hot-drawing of the

hollow billet, reducing its wall thickness and elongating it into the necessary tube lengths.

Second, the plate-cupping process, which consists in taking a circular sheet or plate, and cupping it while hot in a series of operations until it has been formed into a shallow cup of approximately required diameter, then elongating the shallow cup by a succession of hot-drawing operations on a solid mandrel through a succession of dies, thereby reducing its wall to the desired thickness. This process is generally used for tubes of large diameter.

Third, the cast-hollow-billet process, where a cast-steel, hollow, tubular blank is used as a basis of manufacture, which is elongated and reduced in wall thickness by rolling over a plug or mandrel or by hot-drawing operations as previously described.

Fourth, the piercing process, which starts with a round, solid billet, which is passed through angularly-disposed rolls, giving a high rotating speed and a slow advancing movement to the billet, forcing it over a pointed mandrel, increasing its length and changing it from a solid to a tubular form. This is by far the most important of the seamless processes, and, notwithstanding that it is a comparatively new development in the art, it has made great progress and overshadows all other seamless processes in commercial importance. The piercing method is commonly called the "Mannesmann Process," named after the inventor, a German engineer, who made the basic discovery that the cross-rolling of a heated round bar of steel produces a rupturing of the material along its center line, and a tendency to form a hole along its longitudinal axis. This discovery was accidental; it came about in the following manner: Mannesmann was a manufacturer of tool steel, and one of his customers complained about certain bars of steel purchased from Mannesmann for special purposes being out of round, and not having smooth enough surface finish to meet his requirements. In order to overcome these objections Mannesmann built a cross-rolling machine with slightly oval-

shaped rolls, and arranged them one inclined to the other, and so assembled that when rotated they would hold the bar midway between the rolls under pressure, and rotate the bar by frictional contact, at the same time give it a forward as well as a rotating movement. This finishing operation, much to Mannesmann's satisfaction, produced a very round bar with smooth finish; but to his great surprise, when the customer came to use it he found the steel much inferior in quality to the bars formerly furnished. Mannesmann was greatly mystified at this, as the grade of steel used was the same in both cases. In his efforts to ascertain the cause of the failure of the material, he tested some of the same lot of rolled bars that had not been run through the cross-rolling machine, and found the quality of them up to the original standard. He then started to experiment with his new machine to find out the reason for this, giving different inclinations or angularities to the rolls, so as to increase or decrease the forward feed, and was very much surprised to find that when the angularity was reduced to a minimum, which gave a very slow forward feed, with a great deal of cross-rolling, that a small axial hole was produced through the length of the bar operated upon and the metal was cracked radially all round this hole, which accounted for the inferior quality of the steel in the bars finished in the cross-rolling machine. With this discovery as a basis Mannesmann started to find some practical use for it, and naturally thought of its possible application to the making of seamless tubes. He was familiar with the fact that blacksmiths had found that, in twisting iron bars for the manufacture of ornamental iron work, a hole could be opened up along the central axial line of a bar by giving it a severe spiral twist when hot, and he redesigned his cross-rolling machine, substituting conical rolls instead of slightly oval rolls, so as to produce a severe twisting effect upon the bar, which he found very materially increased the tendency for opening up the hole in the center. After further experimentation he found that, in order to control the diameters of the hole and the thickness of the wall, it was necessary to introduce an opposing mandrel. Further experi-

ments showed the necessity of redesigning the roll so as to have a converging inlet and a diverging-shaped outlet pass, with a conical-shaped mandrel located therein. With these changes an ideal pierced billet was produced which was rolled into commercial seamless tubes that immediately assumed a position of engineering importance on account of their superior strength and quality.

There has been much gray matter expended in the developing of variations of the rotary piercing process that is of decided engineering interest, and a general review of its history will be found in an able article pertaining to the manufacture of seamless tubing, written by one of your brother officers, Lieutenant J. O. Richardson, and found in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, November, 1911.

The various means employed for reducing or rolling a short, rough, hollow billet into a long, smooth tube with concentric walls was a problem presenting many difficulties which were only overcome by patient study and effort. The equipment for accomplishing this has been so perfected that hot-rolled seamless tubes of excellent quality and a finish that is hard to distinguish from cold-drawn are now commercially produced, and it is believed that this tube will, in time, practically supplant the cold-drawn product for all boiler purposes.

## MATERIALS FOR STEEL PIPES.

After describing in this very general way the different steps of manufacture of pipe and tubes, the next point of interest to the engineer is, probably, the material from which they are made; and, in order that a comprehensive bird's-eye view may be had of this phase of our talk, I have prepared a table that will show the comparative differences in chemical analysis and physical properties of all the commercial varieties of wrought iron, Bessemer steel and open-hearth steel pipe or tubes of both welded and seamless varieties. Attention is called to the low carbon content of about one-tenth per cent. or lower for

PROCESS OF MANUFACTURE

- WELDED
  - BUTT
  - LAP

  MATERIAL USED: WROT IRON, BESSEMER AND OPEN HEARTH STEELS OF EASILY WELDABLE QUALITY.
- SEAMLESS
  - ROTARY PIERCING.
  - PUNCHING.
  - HOLLOW CAST BILLETS.
  - PLATE CUPPING PROCESS.

  MATERIAL USED: ALL GRADES OF OPEN HEARTH STEEL OF LOW MEDIUM AND HIGH CARBON. ALSO VARIOUS GRADES OF DIFFERENT ALLOY STEELS. IRON CANNOT BE USED, AND BESSEMER STEEL WITH ONLY INDIFFERENT SUCCESS.

| GRADES | | CHEMICAL ANALYSIS S | P | Mn | C | Ni | ELASTIC LIMIT IN LBS PER SQ.IN. | ULTIMATE STRENGTH IN LBS PER SQ.IN. | % ELONGATION IN 2 IN. | IN 8 IN. | % CONTRACTION | HEAT TREATMENT |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| BESS. | AVERAGE | .045 | .100 | .30 | .07 | | 37000 | 57000 | | 22 | 50 | FINISHED HOT |
| O. H. | AVERAGE | .035 | .030 | .38 | .10 | | 32000 | 52000 | | 25 | 55 | FINISHED HOT |
| PUDDLED IRON | AVERAGE | .030 | .200 | TR. | TR. | | 27000 | 45000 | | 12 | 30 | FINISHED HOT |
| .17 CARBON | AVERAGE | .035 | .030 | .50 | .17 | | 40000 | 57000 | | 30 | 50 | FINISHED HOT |
| .17 CARBON COLD DRAWN | MAX. | .040 | .030 | .60 | .19 | | 70000 | 80000 | 18 | 7 | 30 | NOT ANNEALED |
| | MIN. | .015 | .010 | .40 | .14 | | 55000 | 65000 | 12 | 3 | 20 | |
| | | | | | | | 65000 | 75000 | 25 | 16 | 45 | FINISH ANNEALED |
| | | | | | | | 50000 | 60000 | 18 | 10 | 35 | |
| | | | | | | | 48000 | 65000 | 60 | 28 | 60 | MEDIUM ANNEALED |
| | | | | | | | 35000 | 52000 | 50 | 22 | 50 | |
| | | | | | | | 35000 | 55000 | 65 | 35 | 62 | SOFT ANNEALED |
| | | | | | | | 27000 | 47000 | 55 | 28 | 52 | |
| .35 CARBON COLD DRAWN | MAX. | .040 | .030 | .60 | .40 | | 90000 | 100000 | 15 | | 18 | NOT ANNEALED |
| | MIN. | .015 | .010 | .40 | .30 | | 75000 | 85000 | 10 | | 12 | |
| | | | | | | | 85000 | 95000 | 30 | 18 | 32 | FINISH ANNEALED |
| | | | | | | | 70000 | 80000 | 20 | 12 | 25 | |
| | | | | | | | 65000 | 80000 | 45 | 30 | 42 | MEDIUM ANNEALED |
| | | | | | | | 50000 | 65000 | 35 | 20 | 35 | |
| 3½% NICKEL COLD DRAWN | MAX. | .040 | .030 | .60 | .30 | 4.00 | 100000 | 110000 | 18 | | 32 | NOT ANNEALED |
| | MIN. | .015 | .010 | .40 | .20 | 3.00 | 85000 | 95000 | 10 | | 22 | |
| | | | | | | | 90000 | 105000 | 25 | | 35 | FINISH ANNEALED |
| | | | | | | | 75000 | 85000 | 15 | | 25 | |
| | | | | | | | 60000 | 85000 | 50 | 28 | 50 | MEDIUM ANNEALED |
| | | | | | | | 45000 | 70000 | 40 | 20 | 45 | |

both Bessemer and open-hearth steel for the butt- and lap-welded pipe and tubes, as contrasted with the carbon content of .15 to .40 per cent. in seamless tubes. The low carbon content for welded products is a necessity, in order to insure a good weld at the seam. In the weldless tube, however, no such restrictions pertain, and tubes of almost any carbon up to 1¼ per cent. can be made, when required. The three grades of seamless material shown, known as the .15–.20 carbon, .30–.40 carbon and 3½ per cent. nickel-steel of .20–.30 carbon, are the three standard materials most commonly used for a very large percentage of our seamless product. Practically all of the nickel, chromium, vanadium and silicon alloy steels in their different combinations have been made into tubes for special purposes at one time or another. The tendency of many engineers to use special alloy steels for tubular products for a variety of applications has been very much in evidence the last few years, and, in many cases, I fear without a full appreciation of the possibilities of results that could be obtained by intelligent and scientific treatment of the standard grades of carbon-steel found in everyday general use.

The wide range of physical properties of tubes made from these three grades of material produced by different anneals, as illustrated by this table, should convince any engineer of their satisfactory application for almost every purpose excepting something of a very special nature. Should higher elastic limits and tensile strength be required, these can be obtained from this same material, as will be shown a little further along, by the use of a more involved but entirely practicable heat-treatment. This remarkable range of physical properties for the three grades of seamless-steel tubes is secured by the simple means of an open-furnace anneal, where the cold-drawn tubes are subjected to a certain predetermined temperature, which relieves in certain degrees the effects of cold-drawing and modifies the granular structure of the steel in such a way as to give the varied range of physical properties. Careful experiments and standardizing of operations and materials have made it possible to apply this scientific method of anneal-

ing to everyday practical use with the certainty of securing reasonable regularity of results.

The limiting of the materials used in the manufacture to practically three grades of steel is of fundamental importance to the manufacturer, both from an economic and operating point of view, and can hardly be appreciated by those not familiar with the details and complexities of operating a seamless-tube plant; and the desire to restrict the grades of material used to a minimum is not entirely one of selfish interest, as we believe that tubes made from standard grades of steel on which extended practice and experience have obtained have a regularity and general reliability of quality that will be of more value than special grades of tubes that are only occasionally manufactured, with irregularity of quality resulting from heat-treatments not fully under practical control.

My remarks about the lack of appreciation of the possibilities of heat treatment of standard carbon steels have been made to apply particularly to cold-drawn tubes. I believe they could, however, in a general way, be applied with much force to many other articles now being made from alloy steels, the selection or application of which has received little intelligent or scientific consideration on the part of the engineer. As an illustration, I would cite two cases recently coming under my observation of large automobile concerns purchasing alloy steel tubes which, after receipt of same, have to be formed and machined into certain shapes for assembling by the purchaser. These tubes were applied to the machine without heat-treatment, the steel being left in its raw state, which made the tubular part in some respects inferior to the same part made from ordinary carbon steel used by other manufacturers. Great pride was taken in their advertising matter that alloy steel was used in their cars for this particular purpose. Fortunately, in both cases, the designer had provided ample strength of section, so that no trouble resulted; but the use of any ordinary grade of seamless tube would have been entirely satisfactory for this particular purpose, and it could have been applied at about one-half the cost of the alloy steel.

## THE MAKING OF SPECIFICATIONS.

I have endeavored to make my remarks about the physical properties of material quite general. I am now going to talk to you for a while about the making of specifications, and would like to say in advance that I hardly expect that you will concur with me in all the points or conclusions I shall endeavor to make relating to this troublesome subject. My observations and criticisms, however, are made with tolerant and kindly spirit, realizing that in the very nature of things the manufacturer and the engineer will never be able to agree on all of the debatable questions that arise in the drawing of steel specifications; but I will consider my time well spent in making this talk if I can assist you in adopting the habit of fairly considering the manufacturer's side of the case as well as the objects of the engineer in the drawing of steel specifications. My remarks, while intended to apply specifically to specifications for tubes, contain general points applicable to specifications for many other purposes.

It is usually assumed that all standard specifications for steel products call for material of such chemical analysis as is found to be best suited by experience and practice for the purpose for which it is intended. Generally speaking, this may be true; but there is a question in my mind whether tradition and theory have not somewhat influenced engineering opinion as to the importance of adopting restricted and narrow limits for the allowable variation of the different elements, without due regard to the inherent limitations of the process of manufacture over which the steel maker has little or no control. The insisting on requirements just a little beyond the results that can be secured in good everyday practice usually results in hardships to the manufacturer with a proportionally higher cost price to the purchaser, from which no one secures any benefit, as the quality of the steel is not materially improved.

Knowledge acquired during the last few years relating to segregation in steel ingots makes the close specifying of variation in chemical analysis of very doubtful value, unless the manner, location and method of taking drillings for test sam-

ples is intelligently defined. The ladle analysis should give more generally accurate information as to chemical analysis of any heat of steel than drillings taken at random from rolled articles made from the same. The importance of these considerations will be found in a paper read before the American Society for Testing Materials, by Mr. Chas. L. Huston, entitled, "Experiments on the Segregation of Steel Ingots in Its Relation to Plate Specifications": Volume 6, 1906.

A study of this paper shows the futility of demanding from the steel maker impossible homogeneity of composition, and that an intelligent basis for specifying chemical analysis will never be arrived at until the steel maker frankly acknowledges that segregation will probably be found in all grades of merchant steels, and the engineer, in drawing specifications, accepts the existence of segregation as a fact and stops insisting on the impossible, and provides reasonable and fair methods of analysis and tests based on such conditions. If such an understanding could be arrived at, I feel sure that, instead of its resulting in lowering the quality of the standard grades of steel, it would generally improve the quality and result in a more rational and intelligent application of steel products to their many and varied uses.

Closely related to the preceding remarks concerning segregation is the matter of specifying very low-sulphur contents. Many engineers seem to think that the percentage of sulphur and phosphorus in open-hearth steel should be about the same. Specifications requiring phosphorus and sulphur limits of .04 and even .03 per cent. and under are of frequent occurrence. With the basic open-hearth process this is a practical possibility so far as the phosphorus is concerned, but it has little influence in itself in reducing sulphur contents, and .03 sulphur can only be obtained by very careful selection of melting scrap and pig iron or by abnormally increasing the lime contents of the slag, which latter condition, in the minds of many metallurgists, introduces new features in the chemical reactions of the molten steel that probably counteract any improvement in quality that would be expected in the steel by reducing the sulphur

contents to this low percentage. If a low-sulphur content is not obtained on account of the freedom from sulphur in the original pig iron and scrap, it is a debatable question whether the elimination of this impurity by the fluxing action of an increased limestone content of the slag materially improves the quality of the steel. I would like to emphasize this point as strongly as possible, hoping it will have a deterring influence on the tendency of going to extremes in limiting sulphur contents in the drawing of any Navy specifications for which you are in any way responsible.

The great scarcity of low-sulphur ores is a condition that should be recognized as a limitation to the efforts of the steel maker in his endeavor to meet many of the existing specifications. The absorption by the steel during the melting period of sulphur contained in the burning gases in the open-hearth furnace is another handicap to the steel manufacturer in his efforts to obtain very low-sulphur contents. It is well known that all coals carry sulphur as an impurity, and in many cases where very low-sulphur melting scrap and pig iron have been used, the resulting steel contains a higher percentage of sulphur than the original materials from which it is made, the increase being due entirely to the absorption of sulphur from the fuel gases used in melting. In enforcing the limit of .03 sulphur for steel used in the manufacture of tubes it is found that this steel did not weld so easily as .04 and .045 per cent. sulphur, the steel working dry and harsh; but, notwithstanding that this was made very clear to a number of different engineers of railroads which were having trouble in welding safe-ends on locomotive tubes, we were unsuccessful in getting a change in the chemical specification for an increased sulphur content. The raising of the limit to .04 or .045 would not in any way affect the quality of the material for the purpose for which it was intended, as manipulating, physical and service tests after welding do not show any difference in the tubes made from steels with these various percentages of sulphur; and the fetish of low-sulphur content resulted in forcing the manufacturer to furnish tubes made from a steel whose manu-

facture was a hardship and the quality of which was not so well adapted for fabrication or service as a steel of .30 per cent. to .40 per cent. greater sulphur content.

In considering the physical properties we must not lose sight of the probability of segregation of those elements affecting the strength of the material, and sufficient range should be allowed between the minimum and maximum variations to take care of the same, in order that the specifications may be reasonable. Intelligent provision should be made for proper places or location from which test pieces shall be taken. A study of Mr. Huston's paper will clearly demonstrate the importance of this point.

The tendency on the part of many engineers to specify both chemical analysis and physical tests is hardly fair, except insofar as the analysis specified is used to limit the impurities, such as sulphur and phosphorus. If the elements affecting the strength of the material, such as carbon and manganese, are specified, it would appear only reasonable that the purchaser accept the physical properties that would result from the same. On the other hand, if the physical properties are specified and have to be guaranteed by the manufacturer, it would be only fair that he be allowed to use such chemical proportions as would best give these results without being restricted by the specifications. There are some specifications, however, containing both chemical analysis and physical tests, that are intelligently drawn and can safely be accepted by the manufacturer because extended experience and practice for that particular material has shown to a certainty what results can be counted upon; but I have often seen specifications for material, where this experience did not obtain, that the steel maker could not accept without great risk, and consequently had to turn down with the advice that he could furnish the steel to either the chemical or physical tests asked for, but not to both. In most cases of this kind the engineer who has drawn the specifications thinks the steel maker does not know how to make the steel or that he does not care for the business.

It does not occur to him that possibly the specification is unreasonable or perhaps unfair.

A remarkable illustration of a specification demanding a high-grade product without calling for either chemical analysis or physical tests is the specification for boiler tubes of the Bureau of Steam Engineering, Inspection of Material, Part I, Steel and Iron, pages 31 and 32. This specification, as recently revised, is unique in that each individual article has a manipulating test made from coupons cut from each end of every tube that is an excellent general certificate of the grade of material and workmanship of every individual tube; and other manipulating tests made on lots of 100 tubes, which definitely determine the suitability of such material for boiler-tube purposes.

This rational method of drawing a specification was arrived at after a number of revisions and thorough consideration of the practical object of this specification; and after many years of experience in the manufacture of Navy tubes, I do not hesitate to say that the specifying of chemical analysis or physical tests would not in any way improve the quality of these Navy tubes. A quarterly statement is furnished the Bureau of the ladle analyses of all heats of steel used on Government orders, and reasonable limits as to composition, or variation of analysis, are allowed. The Bureau is to be commended for the rational and practical basis used in drawing this specification, and it is unfortunate that some such similar specification is not generally adopted for the purchase of all boiler tubes. But many builders of marine, locomotive and some stationary boilers adopt, in their specifications, very close and sometimes unreasonable chemical and physical requirements, in their efforts to secure a high-grade tube, that are unwarranted burdens to the manufacturer without being a true measure of the quality required, and result in no practical benefit to any one.

It is appreciated that it would be impossible for all specifications of iron or steel articles to be drawn on a like or similar basis as the Bureau specifications for seamless tubes, as the

very high factor of safety existing in boiler tubes renders the question of strength comparatively an unimportant one. But I believe there are many steel specifications now calling for very severe physical tests and difficult chemical requirements that could be revised to advantage, omitting the chemical analysis, excepting perhaps as to the limitation of impurities, and physical properties, and substituting for them intelligently prescribed manipulating tests, which would result, on the whole, in a grade of material better suited for the practical requirements for which it is intended.

I am aware that such a suggestion will, by many, be considered reactionary, and be thought to open the way for passing inferior grades of material; but an intelligent inquiry into the facts as to the general probability of segregation in all steels, the influence of such segregation upon the physical properties, the influence of the different finishing temperatures of the rolled material on the physical properties, and the practical working variations of analysis required for normal mill operations of one heat from another of the same grade of steel, will convince any intelligent investigator that there are many valid arguments that impeach the scientific basis of many of our commonly-used steel specifications.

In commenting on the tendency of many engineers to go to extremes in specifying chemical analysis and physical tests I would also like to call attention to the habit of using extreme and impossible qualifying and contradictory terms in the specifications. We often have specifications asking for soft-annealed tubes *free from scale;* this is practically impossible, as a tube cannot be soft annealed unless heated to a temperature that will produce scale. Or, again, for tubes *perfectly straight;* which is another practical impossibility. Or, cold-drawn tubes *free from all scratches;* which is also outside of the range of practical possibilities. The Bureau of Steam Engineering specifications for steam and water pipe call for *perfectly homogeneous steel,* and pipe that is *perfectly round.* The manufacturer is continually calling attention to points of this kind in specifications that are submitted to him for com-

ment, but the greatest difficulty is encountered in having rational corrections made, as the engineer drawing the specifications believes that if any change were made it might possibly admit tubes that were not of the highest grade of manufacture; and, while in most cases a practical interpretation of the meaning of such impossible terms is taken by the inspectors, in many cases, with new or inexperienced men, they have led to much annoyance, delay and expense before the inspector gains sufficient experience to intelligently interpret the practical meaning of the specification, which should have been made clear in its original writing.

## MILL INSPECTION.

Referring briefly to the matter of mill inspection and inspectors, I would like to make a few observations from the manufacturer's point of view on this subject. The most important qualification for mill inspectors is experience. This coupled with good practical judgment is the most desirable combination, both for the manufacturer and the purchaser. The inspector must have tact, and know how to handle himself about a mill, in order to secure the best coöperation from the mill force in expediting his work. He must be fair and demand absolute loyalty and honesty from employes detailed to assist him. He must maintain the dignity of his position, and command the respect of the mill men with whom he is brought in contact. It might be argued that such qualifications could only be secured from high-priced men, but I am convinced, however, that, by intelligent selection, the services of such men may he had at a reasonable salary. If anything other than the regular mill inspection under the supervision of the manufacturer is required, it is certainly worth doing well, and money intelligently spent for superior mill inspectors is a very good investment and more than pays its own way. The chief inspector should be responsible for the selection and the direction of the activities of all mill inspectors. A system should be established for the proper division of the work with daily records of the condition of each order. This is necessary to

avoid congestion at times when rush deliveries are required. A reasonably accurate knowledge as to the number of different kinds of tubes that can be properly inspected by one inspector in one day is necessary so that the mill inspector will not be asked to do more than is consistent with careful work.

I have frequently seen inspectors compelled to hurry, so as to make quick deliveries, to an extent which prevented the necessary time and attention required for careful inspection. Sometimes the lack of a sufficient number of inspectors is responsible for this condition, and inefficiency pertains to practically all work done. This results in the mill inspectors gradually becoming more or less careless, and leads to the contracting of bad methods that are very difficult to change or correct.

I am making these observations at the risk of their being considered trite or elementary by those who have had experience in the responsibilities of chief inspectors. They are just as important, however, as the remarks previously made relating to specifications, and perhaps more so in the end, because if tubes are inspected by a good inspector it is reasonably sure that they will be first-class, providing the specification is a good one; but, on the other hand, if a poor inspector has charge of mill inspection, no matter if the specification is a good one, there is no certainty that his carelessness or inefficiency will not result in the passing of inferior material, which affects detrimentally both the manufacturer and the user.

## APPLICATION OF TUBULAR SECTIONS TO MACHINE DESIGN.

In taking up the second part of the subject matter of this talk, which I may designate as the application of tubular sections to machine designs, I wish to make it clear that I do not pretend to be a designing engineer in any sense of the term. I have, however, had considerable experience in the manufacture of specially-formed tubes for application to a large variety of different types of machines, and have in that way become familiar with some of the principles underlying such applications, more, however, from a practical than a theoretical point of view; and I will endeavor to develop some

of the fundamental features of application and manufacture that I hope will be of practical interest.

The desirability of the application of tubular forms to different parts of various machinery, which gives increased strength and rigidity and permits decreased weight, is so obvious to you all that it is not necessary to go into the elementary reasons and arguments for such construction. Notwithstanding these facts, however, designing engineers have, generally speaking, been slow to avail themselves of the possibilities of tubular construction, and the tube manufacturers have, in many cases, had to take the initiative in suggesting and working out such applications in their effort to increase the market for their products.

One reason for this is probably due to the fact that the manufacture of seamless tubes is comparatively a new art, and it is only within the last few years that there have been available seamless tubes of a large variety of sizes with various wall thicknesses generally suitable for such purposes.

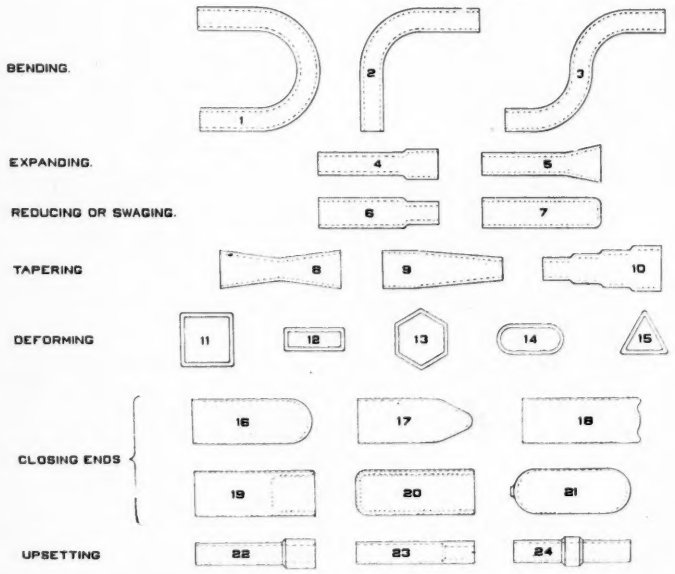
The application of welded tubes and pipe for such purposes is quite restricted on account of the weld being unable to withstand many of the manipulating operations necessary for forming, and the further fact of the limitations of the grades of material available for soft, easily-weldable steel.

The numerous requests from engineers and mechanics for information relating to the mechanical properties of tubular sections made it apparent that there was very little data in the standard engineering handbooks that was of everyday practical use to the designing engineer. The National Tube Company engaged Professor R. T. Stewart, Dean of the Mechanical Engineering Department of the University of Pittsburgh, to collect all available data and make such research as was necessary to compile this information in a condensed form; and in 1908, after much labor involving considerable time for its compilation, the company issued a booklet, entitled "The Mechanical Properties of Seamless-Steel Tubing," for general distribution to tube users. This booklet contains quite a number of entirely new tables and important new information

dealing with the mechanical properties of tubular sections, and places in the hands of the engineer, for the first time, in reliable and condensed form, the necessary theoretical and practical data that make it as easy for him to apply, in designing, tubular sections of many varieties as any of the generally used solid sections of iron or steel. This booklet is now being revised and will be reprinted in the near future with additions of new and interesting data and tables, all bearing on this subject.

The issuance of this booklet on mechanical properties of seamless tubing was very much appreciated by engineers, and doubtless acted as a stimulus in increasing the application of seamless tubing for mechanical purposes. But the expansion of the use of such tubes, notwithstanding our efforts, did not come up to expectations, and frequently, when brought in contact with engineers designing various machines, we found that, while they were desirous of using tubular construction, they possessed a very limited knowledge as to the possibilities of the various shapes and complex forms into which a straight seamless tube could be fabricated. The probable reason for this is the lack of familiarity or experience in the manipulating or forming operations that can be applied in order to secure different varieties of shapes and sections. All good designing engineers are equipped with general practical knowledge accumulated through years of experience, and, in designing, they have a general idea as to how a certain shaped forging could be made, or how a particular shaped casting that they expect to utilize for different parts of any particular device that is under consideration would be molded or machined. But the manipulating and forming of tubes into complex shapes with different sections is comparatively new, and there seems to be a hesitation about adopting or suggesting the use of tubular parts that they are not certain can be made and secured without trouble or at a reasonable expense, and our chief object in dealing with this part of our subject is to make clear some of the possibilities of the different forming and manipulating operations so that the engineer can take the initiative in sug-

## Diagramatic Illustrations Of Mechanical Manipulations.



gesting to the tube manufacturer the possibilities of various adaptations of tubular parts for different machine designs.

The illustration that I will now show you contains some of the mechanical manipulations, in diagrammatic form, that are made on tubes, and its object is to bring before you in one illustration, in an elementary way, quite a number of the possible manipulations commonly used on tubes, showing simple forms with which you are all doubtless more or less familiar.

Figures 1, 2 and 3, illustrating different shapes of bends, are of common everyday occurrence, and the same might be said of figures 4 and 5, illustrating expanding operations; also the same observations could probably be made of figures 6 and 7, illustrating reducing or swaging operations. If, however, we make certain combinations of these different simple manipulations we produce more complex forms, and combining the operations illustrated by figures 5 and 6 in one tube, we would have a form swaged at one end and flared at the other. This type could be combined with any of the three bent forms shown, and produce a more complex form. The operations on figures 8, 9, 10 could be combined with operations on figures 4 or 5 for the whole or part of the length of the tube, or this same combination could be made with the operations illustrated by figures 6 and 7. These different combinations of the various operations made on the same tube would produce a variety of types and shapes of a possible application for which the simple forms would not be adaptable.

All of the preceding figures from 1 to 10 have shown illustrations of tubes of round cross sections, but figures 11 to 15, inclusive, show some of the deformed shapes into which round tubes can be changed, without much practical difficulty, for the whole or a portion of their length, thereby producing more complex forms in their cross-sections of area, made necessary by the particular requirements of the design to which they are to be adapted. As an illustration, the operations on figure 4 and figure 11 could be combined, making the expanded portion of the tube square and the remainder round; or by reversing this order, the expanded portion could be round and the re-

maining portion of the tube square. Or another combination would be by combining operations illustrated by figure 10 with that of figure 11; the largest diameter could be square and the remaining step-tapered portions round.

It will readily be seen that a great variety of combinations can be made, giving a very large number of varied shapes, with different cross sections susceptible of adaptation for various mechanical purposes.

Figures 16 to 21, inclusive, illustrate different types of closed ends that have been found of great practical value in application to formed tubes, both in their straight and manipulated forms.

Figures 19 and 20 are shapes that have been used, showing the more complex forms of closed ends required for special purposes. Particular attention is called to figure 20, illustrating an integral hollow-walled cup or tube closed at one end and requiring an ingenious method of manipulation in its manufacture.

Figure 21 is an illustration of a tube closed at both ends, and used for a float in a water-regulating device. While this is a comparatively simple form, it is of decided practical value in its adaptation to various mechanical purposes.

Figures 22 to 24, inclusive, show diagrammatically three common types of upsets often applied to straight tubes. Figure 22, you will note, is an exterior upset of one end. Figure 23 is an interior upset, and figure 24 is an upset in the middle of a tube. The possibilities of combinations of these three different types is very obvious, and the possibility of adaptation of an exterior and an interior upset for obtaining a heavy thickening of tube at one end is apparent. The combinations of the different upsetting operations with many of the preceding figures is of great practical importance.

It is readily seen that in expanding or flaring tubes, as illustrated by figures 4 and 5, the wall thickness of the expanded portion is reduced; this is sometimes decidedly objectionable, and, to overcome it, the tube is upset preparatory to expanding. Or, again, the expanded portion might be required to be

of heavier thickness than the original gage of the tube, and in this case, before expansion, the tube would be upset still more in order that sufficient cross section of area may obtain in the finished product.

Many of the simple manipulations have been omitted in these illustrations which would further multiply the possibilities of combinations and make possible a further increase in the number of varieties that could be obtained. A sufficient number of primary importance, however, have been called to your attention to illustrate the possibilities of the great variety of forms, both simple and complex, that can be obtained by intelligently arranging the different operations in the required sequence. The means adopted and the tools required for these different manipulations are of interest. We have had prepared a number of lantern slides, illustrating this step-by-step operation, showing the tools in place for performing the different operations, and I am going to ask our Superintendent, Mr. Holinger, who has charge of this particular work for our company, to explain these in a brief way. They show the successive operations, together with the tools used in manufacture, from the original straight tube to the finished form, in some of the interesting tubular articles of manufacture.

In the illustration just shown we have not taken up the matter of formed tubes made from a plate or a solid billet, which are often used for certain purposes. The variations of detail in the manufacture of special tubular articles can only be gone into in a very general way in a talk of this kind; but it is believed that the illustrations that will be shown and described by Mr. Holinger will give you a very good general idea of the whole subject without further explanation by me, and I am now going to introduce him, and ask him to explain his illustrations, which I feel sure you will find interesting.

On the plates to follow will be shown a few selected articles made from seamless tubing; also the making of seamless tubes or cylinders from plates, together with articles other than tubes made directly from the plate; and a few specimens of hydraulic forgings produced from a solid billet.

*Plate No. 1.*

Here is a seamless tube, upset, expanded and flanged, as shown in figure 8; figures 1, 2 and 3 are upsetting operations, where the walls of one end of the tube are thickened or upset by successive operations; the tube is held stationary in a die while a ram or punch is forced against the end of the tube, staving it up until the required amount of material, as indicated in figure 4, has been gathered. Figures 4, 5 and 6 are flanging operations, showing the successive stages required to produce the necessary thickness and diameter of the flange. Figure 7 is the flattening operation, where the metal is finally spread out so as to fill the outside of the flange.

## FLANGING OPERATIONS.

FIG. 1.

FIG. 2.

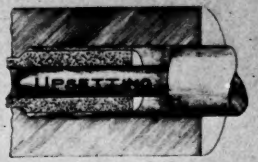


FIG. 3.

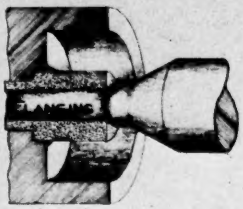


FIG. 4.

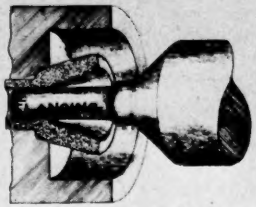


FIG. 5.

FIG. 6.

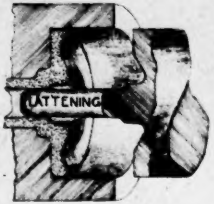



FIG. 7.

FIG. 8.

PLATE I.

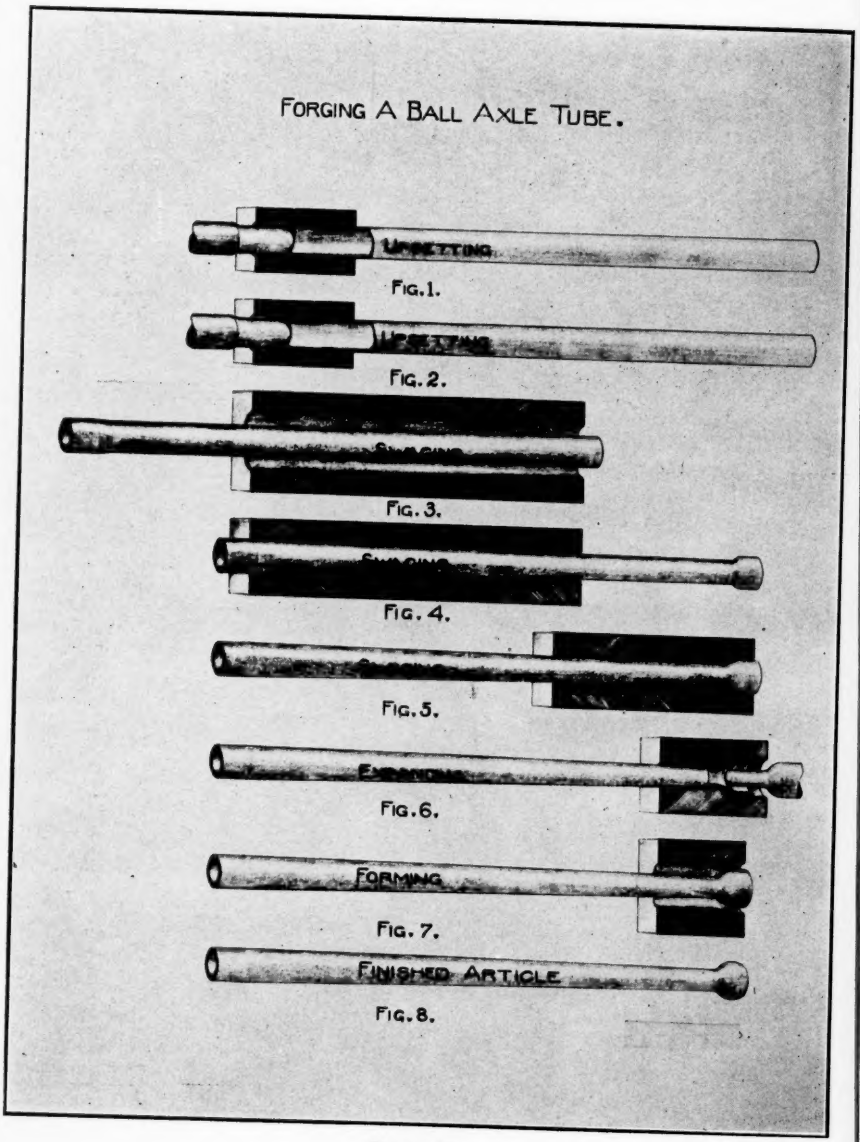

FORGING A BALL AXLE TUBE.
UPSETTING
FIG. 1.
UPSETTING
FIG. 2.
SWAGING
FIG. 3.
SWAGING
FIG. 4.
FIG. 5.
EXPANDING
FIG. 6.
FORMING
FIG. 7.
FINISHED ARTICLE
FIG. 8.

PLATE II.

*Plate No. 2.*

This is another specimen showing the forging of a ball at the end of a tube, besides tapering the tube itself. Figures 1 and 2 show the upsetting operation of the ball end of the tube. The ball on the end of this tube is to be of larger diameter than the tube itself, and therefore the original tube must be upset or thickened in the walls so that in the subsequent operations this end can be expanded to its required diameter and still retain the same thickness of wall as the original tube. Figures 3, 4 and 5 show the swaging or tapering operations which are accomplished by a steam hammer, the anvil of this steam hammer being one-half of the swaging die and the hammer proper the other half. The tapering operation consists simply of feeding the tube in through this die, at the same time turning it while the upper part of the die is working rapidly through the action of the steam hammer.

When long tubes are tapered two or more sets of swaging dies are employed, one set of dies always overlapping the other in the work, so as to prevent any breaks in the true taper of the tube. Figure 6 shows the expanding operation of the ball end, which is accomplished by holding the tube in a split stationary die, as indicated, and with a punch expanding the end to the proper diameter. Figure 7 shows another hammering or swaging operation in a set of dies, in which the ball is rounded up so as to conform to the shape required in the finished article, figure 8.

*Plate No. 3.*

This plate illustrates the tapering and flanging of a tube, figures 1 and 2 showing the upsetting operation of the flanged end, which is necessary in order to obtain the proper amount of metal required to produce the flange; figures 3 and 4 showing the flanging operation itself.

This flanging operation is much simpler than the one illustrated on plate No. 1. Figure 5 shows the tapering operation in hammer dies, with which we became familiar in plate No. 2. Figure 6 shows the finished article, which, in this case, happens to be a flanged and tapered automobile rear axle.

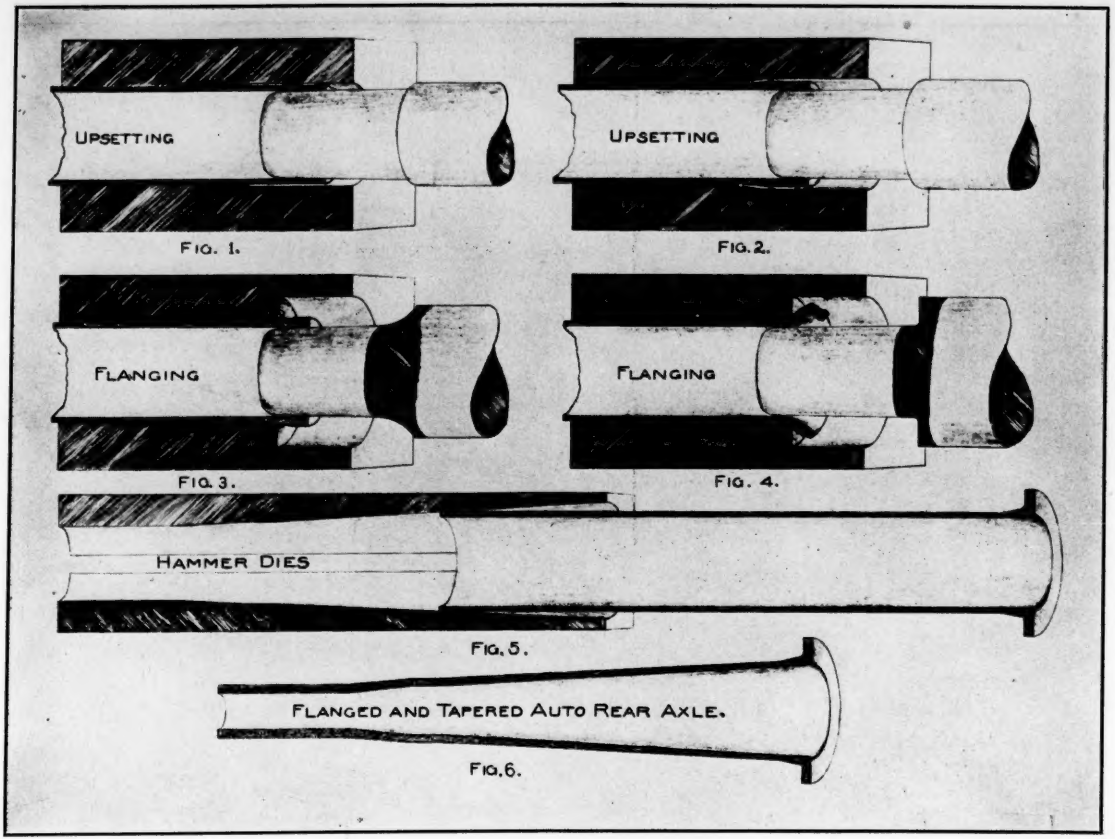

UPSETTING
FIG. 1.
UPSETTING
FIG. 2.
FLANGING
FIG. 3.
FLANGING
FIG. 4.
HAMMER DIES
FIG. 5.
FLANGED AND TAPERED AUTO REAR AXLE.
FIG. 6.

PLATE III.

SWAGING AND UPSETTING OPERATIONS.

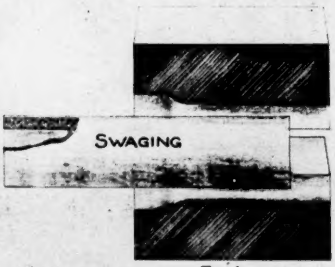



FIG. 1.

FIG. 2.

UPSETTING OPERATION

FIG. 3.

UPSETTING OPERATION COMPLETED

FIG. 4.

FINISHED ARTICLE

FIG. 5.

PLATE IV.

*Plate No. 4*

This plate shows another variation of upsetting and swaging. Here, however, the operations are reversed. The swaging, or reducing of the diameter of a portion of the tube, is done before the upsetting. The tube has three different diameters, which have to be produced with different wall thicknesses, and the most economical method of producing intricate articles is always adopted. The swaging operation, with the subsequent operation of upsetting, brings out in sharp contours the shape required in the finished article.

*Plate No. 5.*

We have here a machine which has lately been introduced into the seamless-tube industry, a spinning machine. The idea of spinning metal is old, but the adoption of the spinning process in the seamless-tube business, for converting tubes into various articles, is new.

In this machine the open end of a tube can be closed by spinning, and also perfectly welded in the same operation. The process is very simple. The machine itself is a sort of lathe with a hollow spindle provided with a chuck for holding the tubes to be spun. Before the tube it put in the spinning machine the end to be spun is slightly heated. The spindle is then rotated at a speed of from 1,200 to 1,400 r.p.m. with the tube. At that speed a blunt tool of self-hardening steel is brought to bear against the tube and swung around the end of it. Little by little the tool is brought forward at each sweep around the end, causing the tube to close up into a half-spherical form. The friction of this tool against the tube creates a high heat, and as the metal is gathered closer and closer towards the center the heat at the center becomes so intense as to melt and close the opening with a perfect weld.

SHOWING THE OPERATION
OF THE SPINNING MACHINE

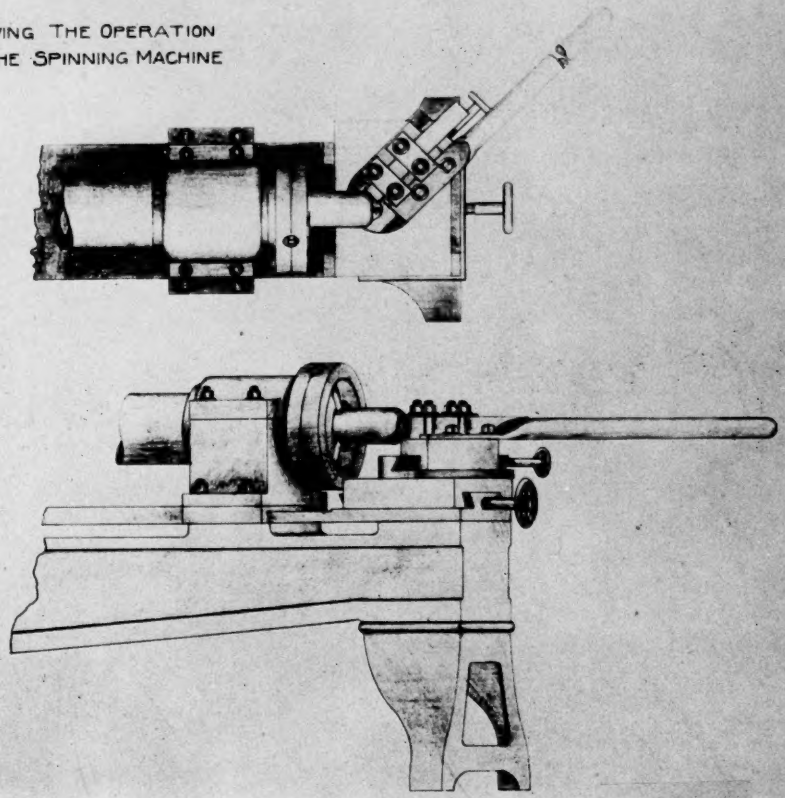


PLATE V.

PLATE VI.

SHOWING FLOW OF METAL DURING SPINNING OPERATION

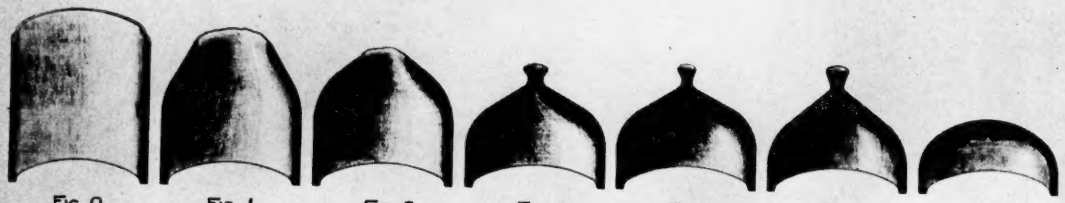


FIG. 0. FIG. 1. FIG. 2. FIG. 3. FIG. 4 FIG. 5. FIG. 6.

*Plate No. 6.*

This plate shows the different stages through which a spun end of a tube passes in this closing and welding operation. Figure 0 is the straight open end of the tube. Figures 1 and 2 indicate how the metal flows during this spinning operation; they show how it thickens up as the diameter diminishes. Finally, in figure 3, the metal has been dragged down to the center but not yet closed; the outside metal of the tube has dragged ahead and formed a button-like protuberance. In figure 4 this protuberance has begun to close up, and in figure 5, when the heat, in the judgment of the operator, is sufficient for welding, the tool is swept across the end of the tube, cutting off this protuberance, and the welding of the center is performed, resulting in a closed tube end as shown in figure 6. This whole operation does not occupy more than 20 seconds. The power required to close a tube 6 inches in diameter by $\frac{1}{4}$ inch wall is about 40 H.P.

*Plate No. 7.*

Here we have an article of manufacture produced by the combined operations of spinning and die-forming. The plain tube in figure 1 is spun and closed as in figure 2. It will be noticed that the closed end in this case is not half-spherical, but pointed. This pointed end is produced by moving the center around which the spinning tool swings beyond the line of the longitudinal axis of the spinning lathe. Figure 3 illustrates another accomplishment obtained by the spinning process. After the end is closed, as indicated in figure 2, the spinning is continued and, gradually moving the spinning tool forward with each sweeping stroke, the metal of the tube walls is caused to flow down and collect at the point; sufficient material is thus collected at the point to fill out the shape required, which shape is obtained in hydraulically-operated forming tools.

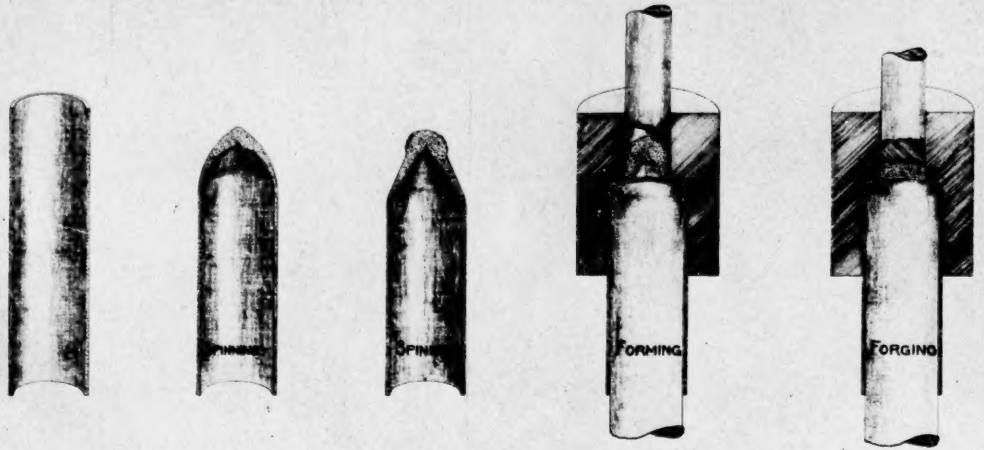



Fig. 1. Fig. 2. Fig 3. Fig. 4. Fig. 5.

SPINNING AND FORGING OPERATIONS.

PLATE VII.

*Plate No. 9.*

We will now briefly describe some phases of the so-called "plate process."

In this illustration is shown all the stages in the manufacture of compressed air flasks, such as are used in submarines and in other places where air or other gases are carried in storage under high pressue.

Figures 1 and 2 show the cupping stage, where the circular plate is cupped, and the cup elongated into a tube by pushing the plate or the cup through dies or rings of proper dimensions. Figures 3 and 5 illustrate the elongation or hot-drawing operations, where the cup produced in figures 1 and 2 is, by successive operations, elongated or drawn by pushing it through a series of dies or rings, each of these rings being a trifle smaller than the previous one through which the tube passes. Thus the Hot-Drawn shell, illustrated in figure 6, is produced. All these operations are performed in hydraulic machines, and the power required to do the work is about 50,000 pounds per square inch of reduction of wall area of tube.

The forming of the open end of this shell into a neck is performed in steam hammers, as shown in figures 7, 8 and 9, with two, three or more sets of dies, depending upon the diameter and thickness of wall of cylinder. Figure 10 is the finished cylinder, now ready for heat treatment, threading of the neck and final testing.

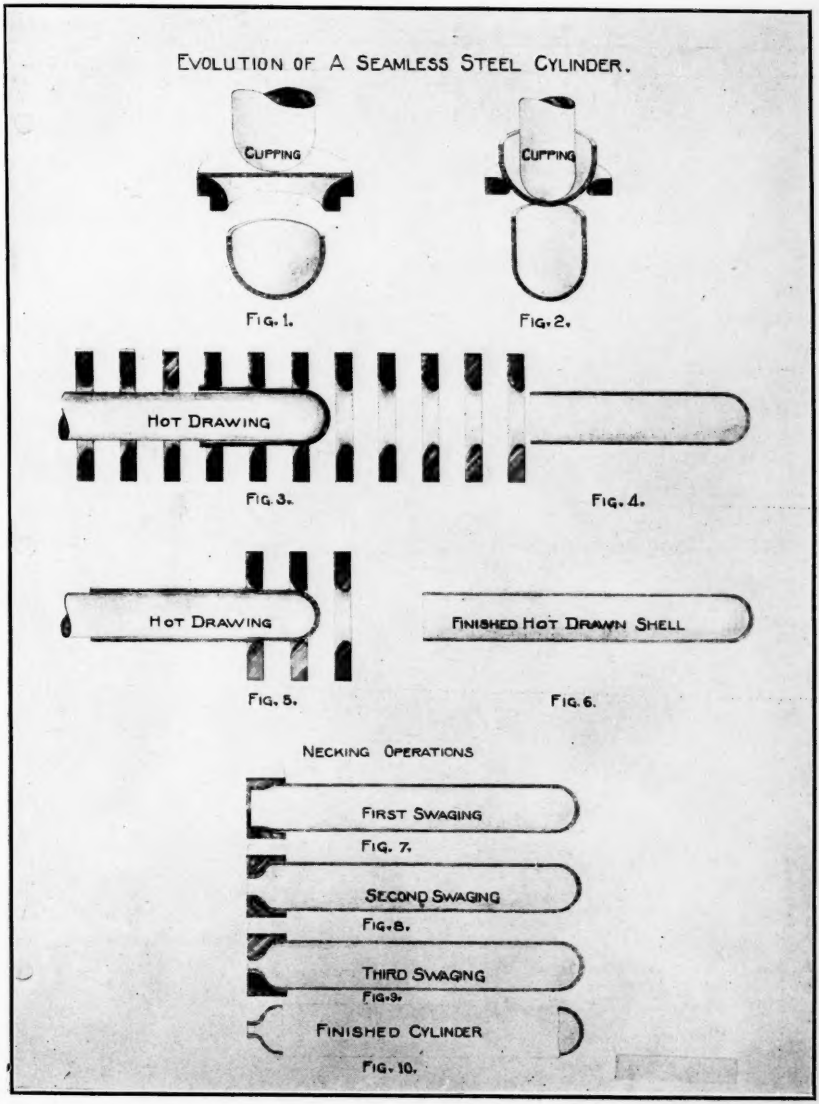

EVOLUTION OF A SEAMLESS STEEL CYLINDER.
CUPPING
CUPPING
FIG. 1.
FIG. 2.
HOT DRAWING
FIG. 3.
FIG. 4.
HOT DRAWING
FINISHED HOT DRAWN SHELL
FIG. 5.
FIG. 6.
NECKING OPERATIONS
FIRST SWAGING
FIG. 7.
SECOND SWAGING
FIG. 8.
THIRD SWAGING
FIG. 9.
FINISHED CYLINDER
FIG. 10.

PLATE IX.

PLATE X.

## SEPARATOR BOWL FORGING

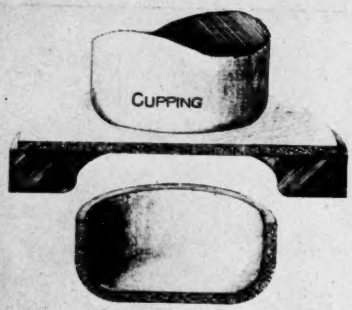



FIG. 1.

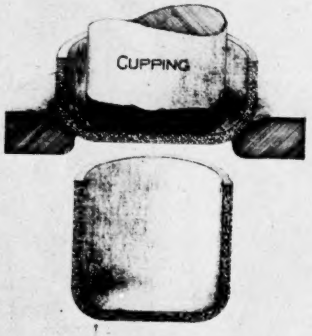



FIG. 2.

FIG. 3.

FIG. 4.

FIG. 5.

FIG. 6.

*Plate No. 10.*

This plate illustrates a forging made from a plate. The usual cupping operations are used in figures 1 and 2. In figure 3 a "squaring" or flattening operation is added, to give the bottom of the cup the proper shape before closing in the top, as in figures 4 and 5, for producing the finished article shown in figure 6.

*Plate No. 11.*

On this plate the manufacture of a locomotive bell is illustrated. Figures 1 and 2 show the cupping operations; figures 3 and 4 a drawing operation, inserted to reduce the wall to proper thickness before the cup is put through the forming operations illustrated in figures 5 and 6. The finished forging, figure 7, is now ready for turning on the outside in order to produce the proper tone of the bell.

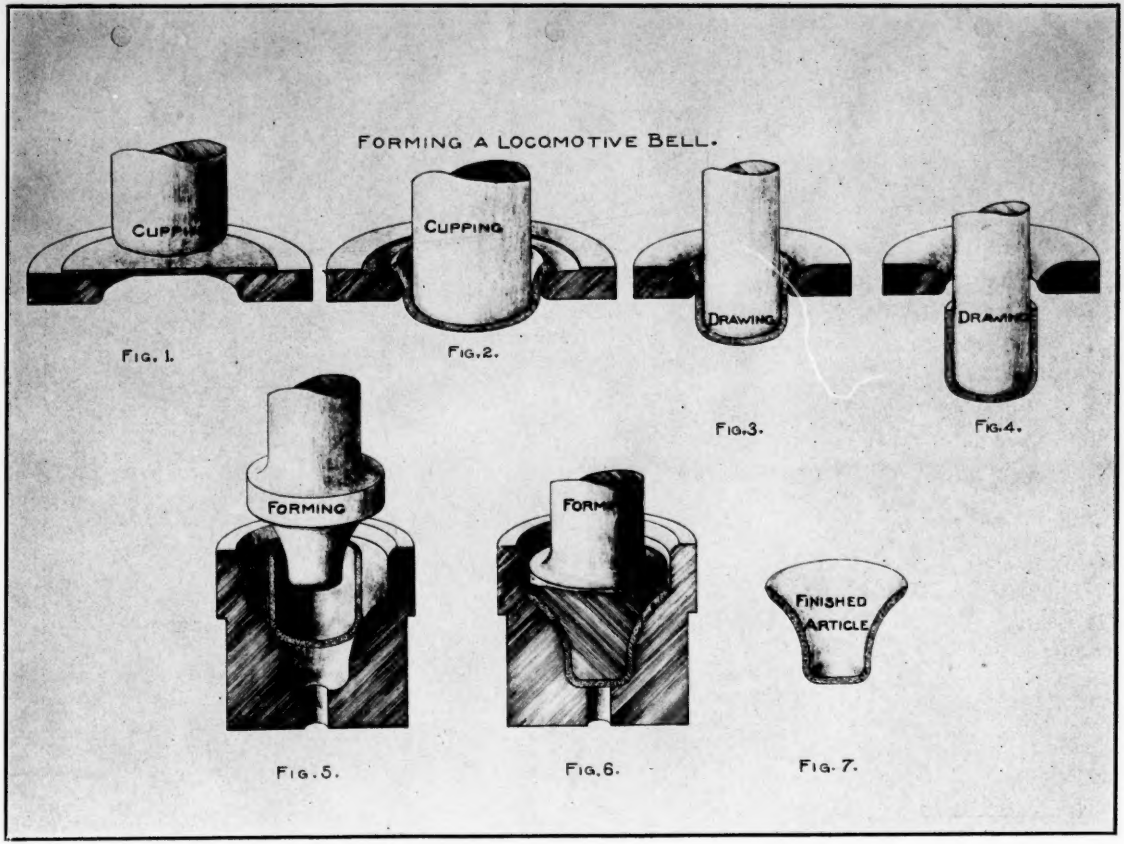

FORMING A LOCOMOTIVE BELL.
CUPPING
CUPPING
DRAWING
DRAWING
FORMING
FORMING
FINISHED ARTICLE
FIG. 1.
FIG. 2.
FIG. 3.
FIG. 4.
FIG. 5.
FIG. 6.
FIG. 7.

PLATE XI.

PLATE XII.

HYDRAULIC FORGING.

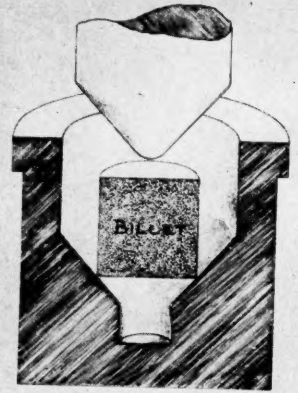



FIG. 1.

FIG. 2.

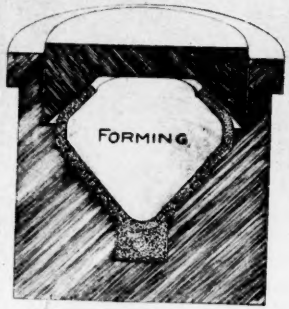



FIG. 3.

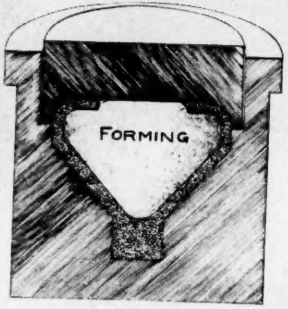



FIG. 4.

FIG. 5.

*Plate No. 12.*

Here we meet with another process, the "hydraulic piercing." Formerly the hydraulic piercing method was quite extensively used in producing seamless tubing, but, after being replaced by other and more economical methods, the piercing process remains only as a means of producing hydraulic forgings. This plate illustrates the manufacture of an article as shown in figure 5, which, in this case, happens to be the forging for the rotating part in a milk-separator. Figure 1 shows the position of the billet placed in the matrix with the piercing punch posing above. In figure 2 the piercing is completed. The power required in this and similar piercing operations is about 50,000 pounds per square inch of piercing-punch area. Figures 3 and 4 show the method of closing in the top of the forging.

*Plate No. 13.*

Here is shown the piercing or forging of a solid-pointed projectile. In this case, however, we have, for reasons of convenience, reversed the position of the tools. In figure 1 the piercing punch is stationary, and the billet containing the matrix is by actuation of a hydraulic piston forced down over the punch, thus piercing the billet. Figure 2 shows the piercing operation completed and figure 3 is the finished projectile forging.

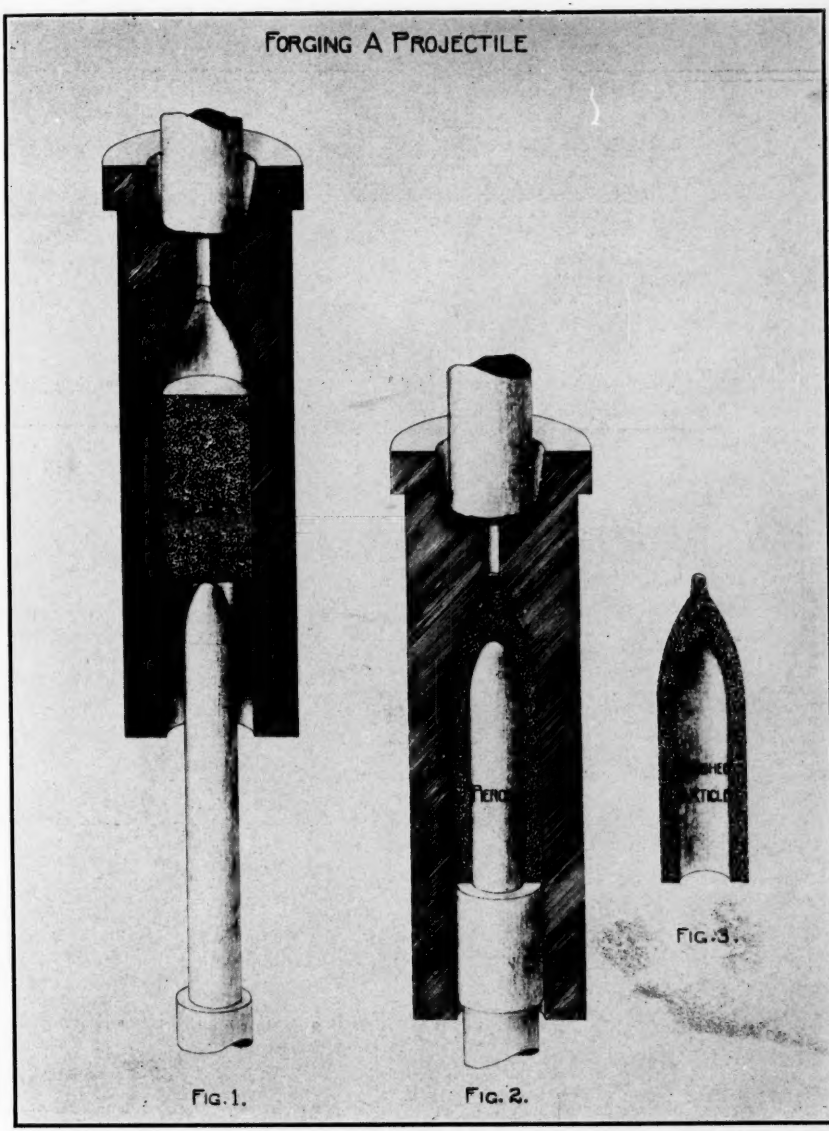



PLATE XIII.

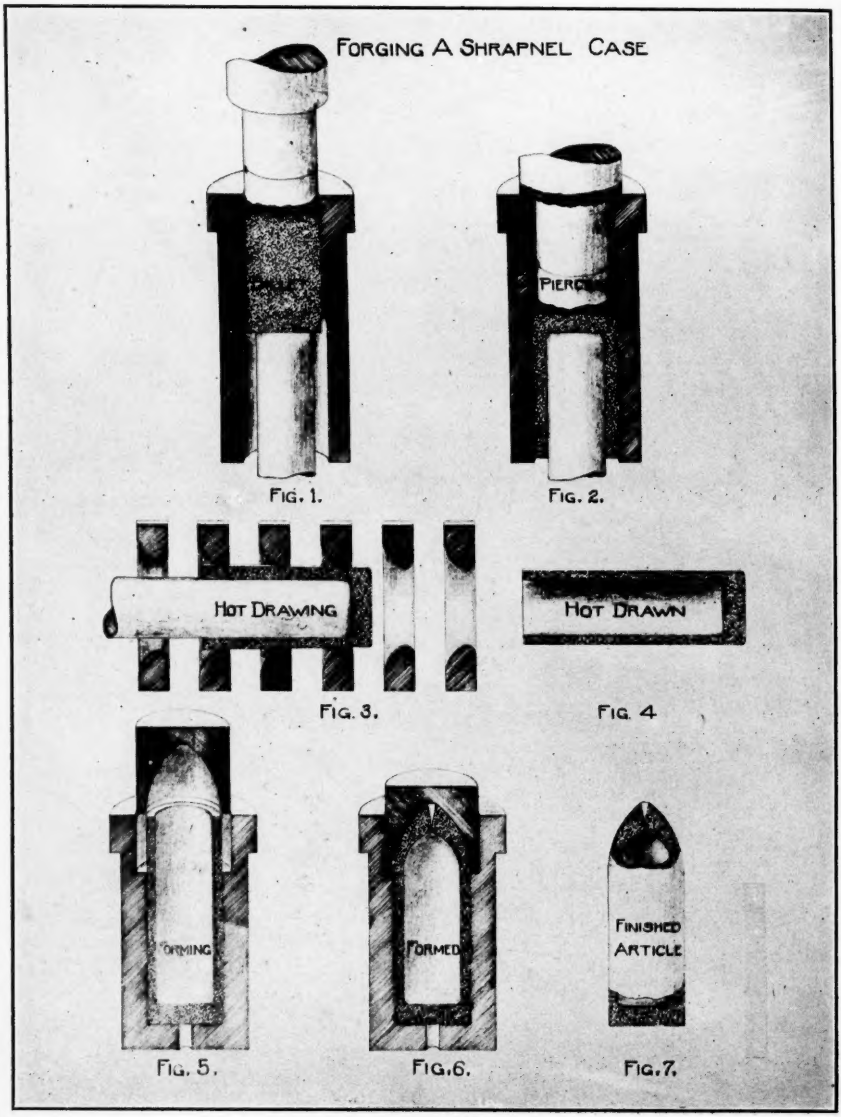



FIG. 1. FIG. 2.

FIG. 3. FIG. 4

FIG. 5. FIG. 6. FIG. 7.

PLATE XIV.

*Plate No. 14.*

To facilitate the work and lessen the friction of the flowing metal we use in making the shrapnel case, shown in this plate, a stationary punch and a stationary matrix; so the only moving part is the hydraulic piston, pushing the billet through the matrix over the punch as shown in figures 1 and 2. A subsequent operation of hot drawing is here required in order to lengthen the shell, which is pierced shorter than the finished shell. Figures 5 and 6 show the closing of the point, with the thickening of the walls required in the finished forging shown in figure 7.

*Plate No. 15.*

This plate shows the making of an open-end cold-drawn corrugated shrapnel case. On account of the pierced cup being very short it is pierced direct, by placing the billet in the matrix as shown in figure 1; the piercing being accomplished as shown in figure 2. The shrapnel is hot-drawn over a punch having the proper corrugations on its surface, as shown in figure 4. The subsequent cold-drawing operation shown in figure 5 elongates the case still more and this cold-draw punch, having also the corrugations on its surface, brings them out sharp and distinct. The finished shrapnel case is shown in figure 6.

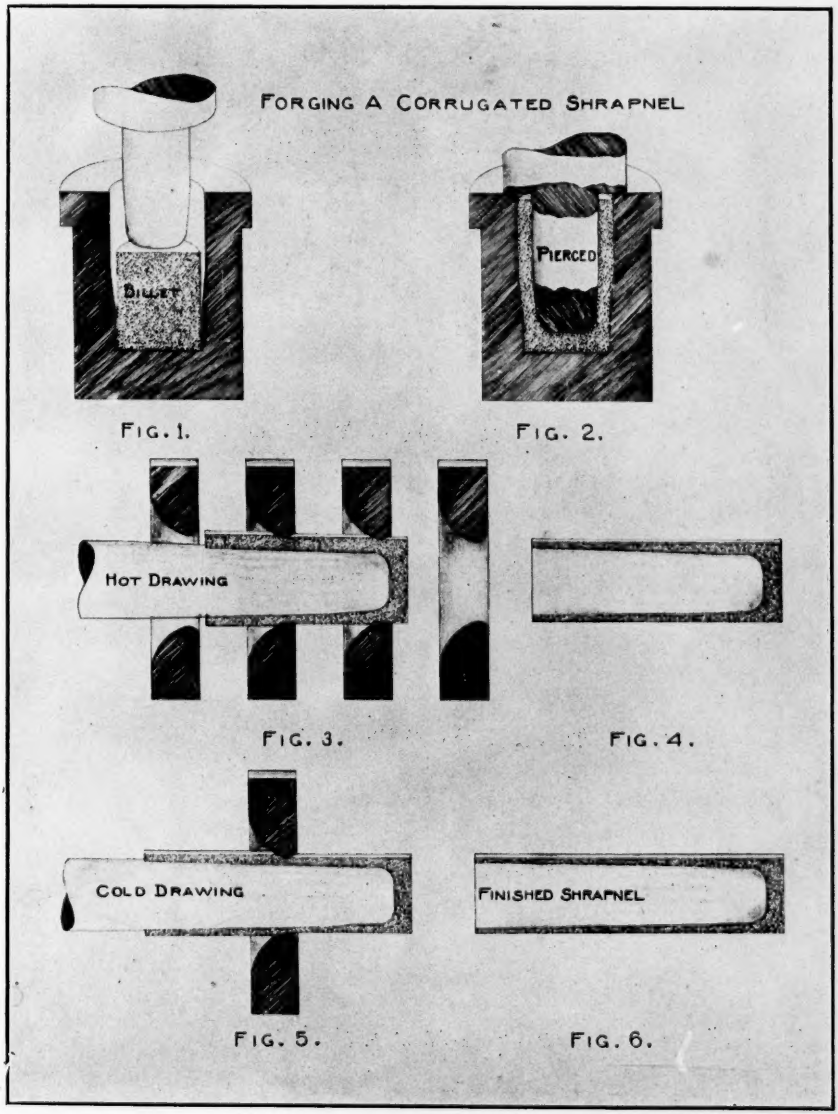



FIG. 1. FIG. 2.

FIG. 3. FIG. 4.

FIG. 5. FIG. 6.

PLATE XV.

SHOWING THE OPERATION OF THE BENDING MACHINE
WITH THE BALL INSIDE OF TUBE

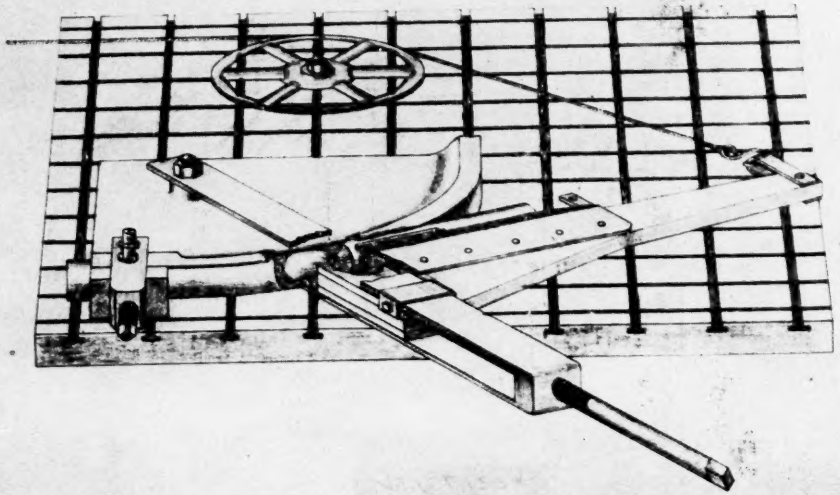


PLATE XVI.

*Plate No. 16.*

The bending of tubes has always been considered a difficult undertaking, especially the production of a bend that would not more or less distort the round section of the tube. This difficulty is partly, if not wholly, overcome in bending the tubes in a simple machine illustrated in this plate. The tube is held in a clamp fastened to a circular form around which the bend is to be made. An arm pivoted to the center of the radius of the bend is provided with a shoe, of half round section, to conform to the circular shape of the tube; this shoe is by means of a screw tightly held down on the tube; a round ball held by a chain having been inserted on the inside of the tube and held there, the chain being fastened to the lever by which the pivoted arm is pulled around the tube form. After the tube has been fastened as described, the arm by means of the lever and wire rope is pulled around the form, bending the tube with it. The ball inside of the tube, following the arm, is always held at the tangent point of the bend, thereby preventing the collapsing or distorting of the tube in the bending operation.

*Plate No. 17.*

This plate illustrates the method used for upsetting ends of large tubing. Such tubes are used for air flasks in torpedo bodies. The operation consists of having a die that fits snugly on the outside of the tube, and, after the tube has been heated at the end to be upset, a hydraulic punch is brought to bear against the heated end, but not upsetting the whole circular surface of the end at one time. The upsetting punch is only four or five inches wide, thus making an impression of its own width, then, being withdrawn and slightly turned, it is again brought to bear against the heated cylinder till it has passed all around. The reason of this piece-meal method, as it were, for upsetting tubes, is on account of the tremendous pressure which would require to upset the tube all around its circumference at once. As it is, it requires a pressure of about 65.000 pounds to make an impression 8 to 10 inches wide on the end of a tube 16 to 18 inches in diameter by about $1\frac{1}{4}$ inches thick.

## UPSETTING LARGE TUBES AND CYLINDERS.

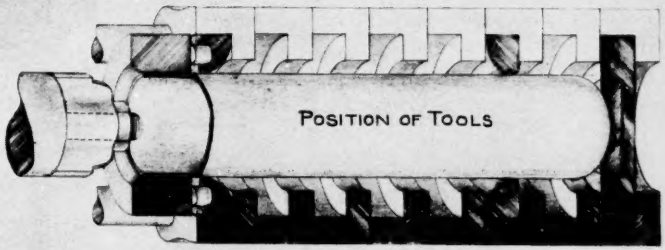



FIG. 1.

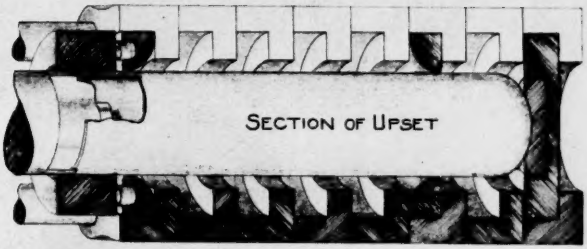



FIG. 2.

FIG. 3.

PLATE XVII.

PLATE XVIII.

## SEAMLESS PIPE JOINTS.

FIG. 1.
TAPER SCREW FLUSH JOINT

FIG. 2.
RE-ENFORCED FLUSH JOINT

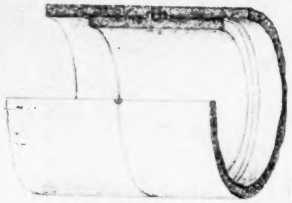


FIG. 3.
SECURED FLUSH JOINT

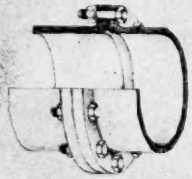


FIG. 4.
SEAMLESS FLANGE JOINT

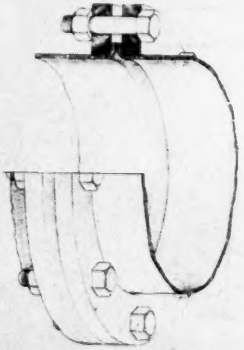


FIG. 5.
LOOSE FLANGE COUPLING

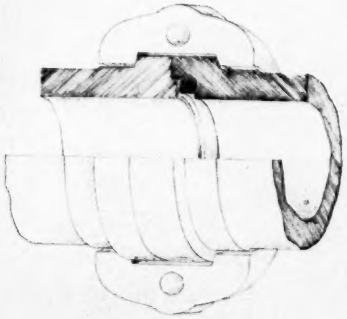


FIG. 6.
HYDRAULIC PIPE COUPLING

*Plate No. 18.*

This plate illustrates a few seamless-pipe joints which are in use today. The plate clearly shows how these joints are made, and requires no special explanation. Figure 1 and figure 2 are threaded joints. Figure 4 is a seamless upset and flanged joint. Figure 5 is a seamless flanged joint, and figure 6 is a seamless upset and leather-packed joint.

# THE INTERNATIONAL RADIO CONFERENCE OF LONDON.

By D. W. Todd, Lieutenant Commander, U. S. Navy, Member.

## INTRODUCTORY.

The need of international regulation of radio-telegraphy was recognized very soon after its availability for commercial purposes became apparent. Its obviously greatest value being that it afforded a means of communication between ship and shore and between ship and ship over ranges many times greater than had been afforded by visual signals, it followed that most of the stations established were shore stations which worked almost exclusively with points outside the limits of the country to which they belonged; and the energy sent out by a station making itself felt with equal intensity in all directions, interference in narrow waters between stations and ships of different nationalities, and between neighboring coast stations, threatened to nullify its benefits to a very great extent. Nations could not make satisfactory use of the new art for naval purposes if commercial activities were not regulated.

In 1903 the German Government invited the principal maritime nations to take part in a preliminary conference in Berlin. Among the propositions which were to be submitted to the governments concerned, as a basis for a future conference, was one requiring all coast stations and ships to intercommunicate without regard to the system of radio-telegraphy employed, and one to insure that stations be so organized as to cause a minimum of interference. The United States was represented by delegates from the Navy, War, State and Commerce and Labor Departments.

In the following year the President appointed an interdepartmental board to consider the whole question of radio-telegraphy as far as government activities were concerned. This board decided that the new art was of greatest value to the government through the Navy Department and recommended that that Department erect a chain of stations along both coasts and on outlying islands, and that the War Department erect such stations as it should require for its own operations. The President approved the report; the Navy's coast system was established; and the War Department installed fire-control stations at various coast defenses, erected an inland system in Alaska, and stations at Omaha and Fort Leavenworth. The board further recommended legislation designed to prevent interference, to prevent the establishment of monopolies, and to place the Government in a position to deal with the international aspects of the subject.

During the following year the draft of a proposed act was prepared by the departments interested, to accomplish the objects mentioned above.

## THE BERLIN CONFERENCE.

In 1906 the first International Conference met in Berlin. The Departments above mentioned were represented. The Berlin Convention was framed, under which the principal European nations have been enjoying carefully prepared regulations for the use of radio-telegraphy for years, and under the terms of which all ship-to-shore communication on our coasts will be brought from December 13, 1912. This convention provides for (1) intercommunication between ship and shore regardless of the make of the radio apparatus; (2) elimination of interference between stations, intentional or unintentional; and (3) making this means of communication a public utility, with reasonable charges for the service.

Before attempting to show the accomplishments of the London Conference it must be recorded that the general excellence of the work done at the Berlin Conference made it unnecessary to make any radical amendments to the Berlin Convention.

Although all parts were thoroughly revised and amended, the additions were of more importance than the changes. The American point of view of the work of the conferences must necessarily differ to some extent from that of Europeans, and some of the reasons will be noted later. It must be borne in mind that the conferences deal only with the commercial side of the question. Naval and military installations are not regulated in any way except that they must be so organized as to cause the least interference possible and must answer calls and assist in cases of ships in distress.

## THE LONDON CONFERENCE.

### *Nations Represented.*

Twenty-nine nations were represented at the conference and, in addition to the representations from mother countries, Bosnia-Herzegovina, Egypt, South Africa, Australia, Canada, British India, New Zealand, and the Dutch Indies and Curacao together, had separate delegations, with voting power equal to that of the mother countries. In all, there were thirty-seven delegations, Austria and Hungary sending separate delegates. All the nations which have subscribed to the Berlin Convention, with the exception of Mexico, sent delegates.

### *The Question of Voting.*

By the Berlin Convention, a government was to be allowed to have one vote for itself and one for each of its colonies and protectorates (not exceeding five), by notifying the International Bureau six months in advance of the meeting of the next conference.

Pursuant to this, Great Britain sought five votes as follows: one each for Canada, Australia, British India, South Africa and New Zealand; Germany, three votes, viz: one for German East Africa; one for the Pacific Protectorate; and one for Togo, the Cameroons, and German South-West Africa. France, five votes, viz: one for French Equatorial

Africa; one for French West Africa; one for Indo China; one for Madagascar; and one for Tunis (Protectorate). Belgium, one vote for the Belgian Congo. Japan, one vote for Chosen, Formosa, Japanese Sakhalin, and the leased territory of Kwantung, taken together. Holland, two votes, viz: one for Curacao, and one for the Dutch East Indies. Portugal, two votes, without stating to which particular colonies or possessions they were intended to apply.

This procedure followed that of the Universal Postal Union, but radio-telegraphy being a recognized instrument of warfare, while postal facilities are developed almost entirely for the benefit of commerce and peaceful intercourse, the granting of votes to colonies was not considered by the United States delegation to the Berlin Conference to be equally applicable to radio and postal congresses. The delegation took the ground that the assignment of a single vote for each nation on all questions was most conducive to the promotion of international amity and confidence and that, at any rate, the principle of the single vote on all matters not strictly commercial would appeal more strongly to the patriotism and self-respect of every nation without colonies. National defense being involved, so also was national existence; and the undue preponderance of voting power of certain leading nations might be a matter of the gravest importance.

The principle of the single vote is not yet established, but Italy, Russia, the United States and Turkey made it plain at the London Conference in more or less positive terms that they considered that their national prestige and dignity would be lowered if they should accept a lesser number of votes than that allotted to any other nation.

During the second day's proceedings Italy made a request for additional votes for her colonies for all questions to be decided at the London Conference, despite the fact that she had not made a request within the time allotted. Russia requested the maximum number of votes at the disposal of any one of the contracting parties, submitting that the number of supplementary votes allotted to the various powers should not

depend solely on the fact that a country possesses colonies, possessions, or protectorates, but also upon the extent of her territory, the length of her coast line, and the number of seas which wash her shores; and submitted that, in case Russia should not be granted equal voting power with any other nation, each nation should be limited to a single vote on all questions of a military or political nature, having reference to the army, the navy, and, in general, to the national defense.

The United States was practically barred from requesting any additional votes for its outlying territories or possessions, on account of the fact that the Senate did not ratify the Berlin Convention until April 3, 1912. The Russian position was strongly supported by the action of the United States delegation in making the same reservation, stating that application for additional votes could not be made within the prescribed time and that radio-telegraphy is of the greatest importance to the United States on account of extent of her coasts and the great distances separating her outlying territories and possessions.

Italy and Turkey made the same reservations as Russia and the United States, and the matter was postponed until nearly the end of the conference. It was practically settled outside of conference, as any extended discussion of the subject in full session might have led to the destruction of the amicable and conciliatory spirit shown by all delegations, noticeably by those of nations not possessing colonies, extended coasts, large maritime interests, or a considerable number of radio stations, whose antagonism to the claims of the leading maritime nations might have been aroused.

The results were that Russia was allotted five additional votes for Russian Central Asia (littoral of the Caspian Sea), Bokhara, Khiva, Western Siberia (littoral of the Arctic Ocean), and Eastern Siberia (littoral of the Pacific Ocean), and the United States, obtained one each for Alaska, " Hawaii and the other American possessions in Polynesia," the Philippine Islands, " Porto Rico and the other American possessions in the Antilles," and the Panama Canal Zone.

Italy at first requested three votes, but consented to reduce them to two by changing a request for one vote each for North Somaliland and South Somaliland to one vote for Somaliland as a whole. She obtained one vote for Erithrea. Following this conciliatory action of Italy, Turkey did not renew her request. Germany declared that the situation had been changed by the increase of voting power of other countries and claimed that she should be allowed two additional Colonial votes, which claim was allowed.

The United States delegation had received special instructions concerning this matter of votes from the State Department, and accomplished the object desired.

COMPOSITION OF THE DELEGATIONS.

The conference being called to revise a convention framed to promote the efficient use of radio-telegraphy for commercial purposes, the greater parts of all delegations were officials and experts of the various post and telegraph Departments, under which are all coast stations and land telegraphs in European countries.

About 41 per cent. were post office and telegraph officials, administrators and engineers. Nearly one-third were naval and military officers, because it was apparent that all European nations recognized the very great importance of the art as an instrument of war. The naval and military element was present to see that no provisions for efficient commercial working should in any way interfere with the arrangements made for offense and defense. One-sixth of the delegates were technical men or men of special attainments in the scientific world. A number of them were directly interested in cables, as one or two of the propositions clearly indicated the belief that radio stations will eventually handle considerable over-sea work between points at present connected only by cables. With several of the delegations were diplomatic officials of high standing to watch matters connected with national policy.

France sent 13 delegates; the United States, 12; Great

Britain, 10; Germany and Russia, 9 each; Italy, 8; Spain and Bosnia-Herzegovina, 6 each; Denmark, Norway and Japan, 4 each; Belgium and Turkey, 3 each; and each of the other nations and dependencies two, or one, delegates.

The United States delegation was headed by Rear Admiral John R. Edwards, U. S. Navy, as chairman, and was made up of three representatives each from the War and Navy Departments, four from the Department of Commerce and Labor, one from the Department of Agriculture and one from the Treasury Department. The Department of Commerce and Labor also sent an attaché, as technical adviser to its delegates.

While most delegations included a number of technical men named as delegates, a few had them named as attachés. Of the largest delegations only certain members were empowered to sign the convention, except in the case of the United States whose delegates were all plenipotentiaries. All signing delegates and others, and attachés were seated together in the hall. Various commercial radio companies were permitted to be represented, and these representatives attended all the meetings of the conference, and of the committees, were permitted to give their opinions and to take part in the discussions, but were not given the right to vote on any matter. This not only insured the practicability of the working rules adopted, but gave the conference the result of practical experience beyond that acquired by the various government officials. Every opportunity was given the commercial radio companies to put forth their own ideas and to guard their interests.

THE CONFERENCE HALL.

The meetings of the conference were held in the hall of the Institution of Electrical Engineers on the Thames Embankment. The seats were arranged in an amphitheater, and the President could see each person from his platform. Seated by the President of the Conference was the general secretary who was the Vice President of the International Radio-telegraphic Bureau at Berne. On each side were the other secreta-

ries and reporters; some from the British Post Office and some from the International Bureau. The delegations were arranged in alphabetical order of the French names of their countries, all the delegates from one country being seated together in one tier. This placed Germany in the front row and the United States and Austria in the second. Nations having representatives from colonies or protectorates had the colonial representatives sitting next to or adjacent to those of the mother country. Representatives of commercial companies occupied seats at the back of the hall. Each delegate or representative had a desk. Besides the main hall, there were a number of committee rooms, smoking rooms, etc., and in the entrance hall was a post and telegraph office through which each delegate had free use of not only the post and telegraph systems of the United Kingdom, but could cable to his home country without charge.

### OPENING OF THE CONFERENCE.

The conference was opened on June 4, 1912, by Mr. Herbert Samuel, Postmaster General. In his welcoming speech he referred to the admirable work of the Berlin Conference, as exemplified by a convention which had worked so smoothly and effectively that, although provisions had been made for the denunciation of the contract by any nation and for settlement of disputes between nations working under it, no occasion had arisen for any nation to exercise its right to withdraw or to seek arbitration. He also referred to the advances made and to the importance of the work of the present conference, especially in the matter of providing additional safeguards for life and property at sea. He closed by requesting the conference to accept as its president, the head of the British Delegation, Sir Henry Babington Smith. This very accomplished diplomat, linguist and administrative expert of Post Office affairs, at one time Secretary of the British Post Office, and a delegate to both radio conferences at Berlin, was accepted with enthusiasm and took his seat as presiding officer on the withdrawal of the Postmaster-General. The remarkable tact, patience and ability of this man was a most

potent factor in smoothing difficulties and in guiding the discussions occasioned by the various points of view of the different delegations.

### THE METHOD OF CONDUCTING BUSINESS.

Certain rules of procedure similar to those followed at the Berlin Conference were proposed by the British delegation and accepted. Among them were: (1) Rules for voting. (2) Duties of the President and Secretaries. (3) Use of the French language for all debates and documents. (4) Publication of minutes. (5) Seating of the delegates. (6) Manner of presenting propositions. (7) Method of adopting propositions. (8) Contents of published proceedings. (9) Appointment of committees and their manner of conducting business, and adoption of the results of their discussions by the Conference. (10) Work of the Drafting Committee and manner of adoption of its work.

There were nine sittings of the full conference in which were settled all the articles of the convention itself. All specially difficult subjects and all the work of the committees to be mentioned hereafter were read, discussed and adopted in their final form. The detail work was delegated by the full conference to three committees, two of which were composed of the delegations of the principal maritime powers in such numbers as to make their work practically that of the conference as a whole. These two committees were the regulations committee and the committee on tariffs. The third committee, called the drafting committee, was composed of the delegates of five countries.

The sittings of the full conference were held principally at the beginning and at the end, while during the greater part of the month of June the regulations committee and the tariffs committee worked sometimes on alternate days, and sometimes each committee held a session on the same day, forenoon and afternoon.

### REGULATIONS COMMITTEE.

This was composed of the delegations of twenty-two countries, among which were all the principal European nations and the United States. The head of the French Delegation, the Director of Research of the Postal and Telegraph Department of France, presided over its ten prolonged sessions, and the tact and ability shown by him in the performance of his somewhat arduous duties won the admiration of all.

The work of this committee was the examination of all the regulations of the Berlin Convention, bringing them up to date and acting on the various proposals offered connected with the operating rules and with questions of a technical nature. The purely technical questions were handled by a special sub-committee, in which practically all who wished took part. As in the case of the main committee, the principal nations were all represented, and took an active interest in the discussions.

### COMMITTEE ON TARIFFS.

This committee, under the presidency of the head of the German delegation, one of the strongest men of the conference and the head of the Post and Telegraph Department of the German Empire, handled all questions of accounting and tariffs. The work of this committee was very important and, as in the case of the regulations committee, the delegations of the leading European countries took part.

The six sessions were attended by the United States delegation, but it refrained from taking part in the discussions on account of the fact that no regulation of rates or accounts had been established in our country.

### THE DRAFTING COMMITTEE.

While the first two committees were deciding on the matters that were to be incorporated in the regulations of the convention, the drafting committee was preparing to bring the various features of their work into accord.

The exact wording of the various regulations adopted were decided upon, repetitions and inconsistencies eliminated, and the various parts arranged under suitable headings. The committee was under the presidency of the head of the Belgian delegation, the Director-General of Telegraphs, and held six sittings. Its work was very arduous and was so well performed that the various articles were adopted by the conference in the form proposed by it at first reading.

## INTERCOMMUNICATION.

One of the first matters to be brought to a satisfactory conclusion was the question of intercommunication between ships.

At the Berlin Conference, the doctrine of intercommunication was strongly advanced by Germany and several other nations, and supported by the United States. Without such a provision the value of radio-telegraphy for commercial purposes would be very much reduced except through the workings of a world-wide monopoly, with consequent artificial and arbitrary restraints upon trade and intercourse. Several so-called "systems" had been developed, but the Marconi Company, through its contracts with the British Government and with the great corporation of Lloyds, sought to stifle competition by having their ships and stations refuse to communicate with any ship or station working apparatus of any other system. This matter caused the greatest amount of discussion in the conference, and was finally settled by all nations subscribing to the principle that all ships should communicate with all shore stations. Great Britain, Italy, Japan, and Portugal, however, did not sign a "final protocol" in which the other nations declared that they would require, in addition, all ships under their flags to intercommunicate without any discrimination. Italy declared herself in favor of the principle but was unable to subscribe to the final protocol until after expiration of her contracts with Mr. Marconi. The principle, based on equity and intended to further the most effective use of the new art, has finally been recognized by all nations. At the second plenary session it was announced that Great Britain,

Italy, Japan, and Portugal had subscribed to the final protocol of the Berlin Convention, to the end that the provisions of this protocol might be incorporated in the convention itself. This announcement was loudly applauded. At a later session, it was announced that the Marconi company had decided to recognize the principle in advance by ordering all ships fitted with its apparatus to communicate with others.

RADIO COMMUNICATION BETWEEN FIXED POINTS.

The proposition to regulate commercial work transacted over land and sea between land stations brought up the subject of intercommunication anew, and there was an extended discussion between Great Britain, supported by Brazil, Canada, and Italy, in favor of not recognizing the principle as applied to fixed stations, on the one hand; and Germany, supported by France, the United States, Roumania, and Hungary, on the other; the last named countries declaring that, while they agreed with Great Britain that it was too early to regulate such radiocommunication on account of lack of experience with that class of work, except in so far as non-interference and answering signals of distress from ships were concerned, it should be clearly stated that stations operating between fixed points should not refuse to exchange radiograms on account of the difference of the systems employed. It appeared that Great Britain wished to avoid any possibility of having to handle messages offered by a high-power station of another country to any of the high-power stations which it proposed to build in Africa and Asia. The matter was decided by a vote of 28 to 20, the principle being sustained by the adoption of the following clause: "However fixed stations which carry out correspondence between land and land must not refuse the exchange of radiograms with another fixed station on account of the system adopted by such station;" and, by the adoption of the following clause, Great Britain's right to control absolutely the business to be handled by her proposed stations was acknowledged: "Nevertheless, the liberty of each country shall remain complete in respect of the

organization of the service for correspondence between fixed points and the decision as to the correspondence to be carried out by the stations appropriated to such service."

PROPOSITIONS.

The various nations which had adhered to the Berlin Convention sent in a list of propositions for discussion six months in advance of the London Conference. These were published in French by the International Bureau at Berne and were translated and printed by the Navy Department in order that they might be studied. Copies of this translation were sent to various commercial radio and telegraph companies and comments were invited. In addition to this early publication by the Berne Bureau, other proposals were advanced by various nations at the opening of the conference and from time to time during the sessions. The Senate having ratified the convention on April 3, 1912, and the President having proclaimed it in May, 1912, no American propositions were published in advance, but they were submitted to the conference from time to time during the sessions. Over two hundred propositions were submitted in all, by various delegations, but many of them were duplicates and represented different expressions of the same ideas by different nations. On the whole the British propositions were those that met with the greatest success. They seemed to be quite conservative and apparently suited the situation in Europe very well.

AMERICAN PROPOSITIONS.

Not having carried on any radio work under the Berlin Convention in this country, and not being a party to the International Telegraph Convention, the American point of view was somewhat different to that of European nations. While they had been carrying on their radio work in narrow waters under the terms of the Berlin Convention with a considerable degree of success, and were using methods of operation analogous to those in use in the European system of land telegraphy,

the United States had not regulated radio-telegraphy in any way, and the methods of operating and accounting used by radio companies in the United States were not standardized on account of this lack of government control. The land lines, as well as the radio stations of Europe are owned by the governments, and all nations use the same operating methods quite different to those used by the telegraphs of this country, but similar to those almost universally used in cable work.

WAVE LENGTHS.

The American delegation took an advanced view of the possibilities of increasing the amount of work in a given locality by allowing a greater freedom in the choice of wave lengths for commercial work, relying on the sharp tuning which it was proposed to obtain by the legislation then pending. This legislation was passed on August 13, 1912, is known as "An Act to regulate Radio Communication."

The American propositions regarding wave lengths were based on the sharp tuning provisions of this act. The American delegation even went so far as to recommend that the limit of the range of commercial wave lengths most suitable for ships be increased from 600 to 800 meters, which would narrow the range reserved for the exclusive use of naval and military installations. The theory was that sharp tuning would so increase the work possible both for government and commercial installations, that the loss to the government would be small and the advantage to commerce very great. It was soon found that there was not the slightest tendency on the part of any foreign delegation to interfere with the governmental radio activities in any way. The unsettled conditions in Europe and the great importance placed on military preparedness by European nations, made them unwilling to entertain any proposition which would upset their present military arrangements in the least. No proposition would be considered by the committees unless it were seconded by another delegation, and this particular feature of the American proposition was not seconded.

The other wave-length feature met with no greater favor, one which would permit commercial ships and stations to use a number of wave lengths between 300 and 600 meters, which might be suitable for general public correspondence in special circumstances, provided only sharp waves were used. The foreign idea about this was that all the traffic of the English Channel, the North Sea and the Mediterranean was being handled on two wave lengths only, that is 300 and 600, had been so handled for several years, and that their experience was that a wide difference of wave length was necessary to prevent interference.

The change from 300 to 600 meters as a standard calling wave length for ships was proposed by the United States and other delegations, 600 meters being a more efficient wave length for ordinary use by ships. It was also made clear that the standard wave length should be used for distress purposes.

The proposition to use a fixed long wave for time signals, weather reports and maritime information sent from certain stations was not adopted.

Vessels of small tonnage will be required to use 300 meters exclusively in sending; other vessels and all coast stations must be ready to use either 600 or 300 meters.

In France, radio signals for navigation purposes are now being sent out from stations called radiophares. The wave length of these stations will be limited to 150 meters, not only to prevent interference with ordinary correspondence, but because the shorter wave lengths are those most efficient for fixing ships' positions. The distance over which this method of fixing a ship's position is efficient is estimated to be about 30 miles at present, and the range of radiophares will be limited to this.

### SHARPNESS OF TUNING.

The proposition that the Conference define exactly the degree of sharpness of tuning allowable for all installations was offered by Germany and the United States, the latter country following the " sharp wave" paragraph of the " Act to Regu-

late Radio Communication." Great Britain, France and Italy declared that this was not practicable at present, and supported the French proposition that the waves emitted should be "as pure and as little damped as possible." This was adopted by the conference, and is merely advisory in character. Spark gaps in aerials, however were forbidden except for distress purposes, when a maximum of interference and radiation is desirable, and for small vessels using not more than 50 watts primary energy.

Germany and the United States were also in accord in recommending that a transmitter be rated by the amount of energy in the aerial, but the definition of the Berlin Convention was retained, the rating being by the energy measured at the terminals of the generator. This is considered unfortunate, as different sets with the same primary energy radiate amounts of energy varying in the ratio of 50 to 1.

The appointment of an international committee to determine upon standard methods of measurements and a standard nomenclature was recommended, but the idea was considered premature.

The United States delegation proposed to define the degree of selectivity of the receiver. This was also put aside and a very general provision adopted in its place, viz:

"Receiving apparatus must allow of receiving, with the greatest possible amount of protection from disturbance, transmissions made with the wave lengths specified in the present Regulations up to 600 meters."

### SAFETY AT SEA.

The conference following so closely the loss of the *Titanic*, it was to be expected that the widespread discussions of the part played by radio apparatus and operators would result in a number of propositions designed specially to prevent or minimize the loss of life in cases of collision. Generally, all regulations tending to facilitate communication between ship and ship and between ship and shore lessens the danger of a recurrence of such a frightful catastrophe. The abandonment of

the principle of non-intercourse between ships fitted with apparatus of different systems, while it was about due in any event, was assured after the disaster. The fact seems to have been established that, although over 800 lives were saved, many more, possibly every soul, on board the *Titanic*, might have been saved through the agency of radio-telegraphy. The events of the night of April 14 and a few succeeding days and nights, are still so clear in most minds that they will be readily recalled by reading the provisions incorporated in the London Convention, and the resolutions adopted at the conference, that were suggested by those events. These provisions were intended to lessen the danger of distress calls going unheard, to prolong the availability of the radio apparatus for distress calls, and to prevent confusion, by the following recommendations and requirements:

The compulsory installation of radio apparatus under the charge of skilled operators on certain ships, and the establishment of an adequate number of coast stations with constant service by each maritime country, were recommended.

The London Convention will require the following:

(1) A constant watch in the radio rooms of certain ships.

(2) Regular hours of "listening-in" on vessels not required to keep a constant watch.

(3) Installation of auxiliary apparatus capable of working independently of power from the ship's boilers for six hours.

(7) Interruption of press or other long transmissions at intervals to permit ships and stations to listen for distress calls on the standard wave length.

(5) Placing the radio apparatus and operators directly under the authority of the captain.

(6) The control of all radio communication in the vicinity of a ship in distress by that ship.

With the exception of specific times for listening required of vessels not keeping a constant watch, the above items were all brought forward by the United States delegation, either by quoting legislation passed or under consideration by Congress, or in special propositions.

## AMERICAN LEGISLATION.

Printed copies of the "Act to require apparatus and operators for radio-communication on certain ocean steamers," approved June 24, 1910, and its amended form approved July 23, 1912; and the "Act to regulate radio communication"; were placed before every delegate to show the attitude of the United States on various propositions. The first act forbidding ocean passenger-carrying vessels of any nationality to leave a port of the United States with 50 or more persons on board without radio apparatus, with a normal range of at least 100 miles, in charge of a skilled operator, was the first legislation ever enacted for compulsory radio installations, and this placed the United States far in advance of other nations in the matter of insuring safety of travel by sea. The act went into effect over eight months before the loss of the *Titanic*. After the disaster, the amended act was passed, extending it to cover any vessel carrying 50 or more persons, whether any passengers are on board or not; applying it to the traffic on the Great Lakes; requiring auxiliary apparatus independent of the main electric plant and capable of working for four hours over a distance of 100 miles; requiring at least two skilled operators and a constant watch on such vessels when underway; and placing operators and apparatus under the direct control of the captain, on American vessels.

The London Conference did not consider itself competent to do more than recommend the compulsory equipping of ships and the establishment of additional coast stations, but passed unanimously the following resolution:

"The International Radio-Telegraphic Conference having examined the measures to be taken with the view of preventing disasters at sea and of rendering assistance in such cases, expresses the opinion that, in the general interests of navigation, there should be imposed on certain classes of ships the obligation to carry a radio-telegraphic installation.

"As the Conference has no power to impose this obligation, it expresses the wish that the measures necessary to this end should be instituted by the Governments.

"The Conference finds it important, moreover, to ensure, as far as possible, uniformity in the arrangements to be adopted in the various countries to impose this obligation, and suggests to the Governments the desirability of an agreement between themselves with a view to the adoption of a uniform base for legislation.

"Lastly, the Conference recommends to the Governments the desirability of establishing in each maritime country a number of coast stations with a permanent service, adequate for the needs of navigation."

METEOROLOGY.

To increase the safety of shipping, an important proposition was advanced for the simultaneous reporting of barometric pressures by all ships in the North Atlantic to Washington or to London or Paris, depending upon whether the ships reporting were west of the 40th meridian of west longitude or east of it. If at a given hour each day, for instance, at Greenwich mean noon, every ship was to observe the height of the barometer and transmit the same to the nearest coast direct or by relaying from ship to ship, using two code words, the first giving her position and the second the reading of the barometer, a pressure chart for the entire North Atlantic could be prepared which would be of inestimable value in foretelling storms. The American delegation proposed to make this compulsory for all vessels, and the American and European Weather Bureaus were to plot the readings and exchange data for mutual benefit. It was pointed out that many storms of great intensity originated in the West Indies or on the northern coast of the United States and travel westward across the North Atlantic, so that the reporting service would be possibly of greater advantage to European countries than to our own. It was found, however, that the foreign administrations were not prepared to go as far in the matter as the United States Weather Bureau, one of the objections being the matter of expense. A proposition of Monaco on the same general principle but not so broad in scope was adopted. By

it each government is left to make its own arrangements for obtaining the barometric pressures, and priority of weather messages is assured by classing them as paid service advices to be transmitted at the expense of the government concerned. This priority of transmission is the feature by which the success of the proposed service is assured.

The convention will require that coast stations be supplied with weather reports and storm warnings of local interest, to be sent to ships on request.

Time signals and weather reports will be sent together, and all ships and stations will keep silent while they are being sent, exception being made for distress calls and important government business.

ROUTE OF MESSAGE.

The situation as regards land telegraph lines being quite different in the United States to that abroad, provision was made for allowing the sender of a message on board ship to designate, without additional charge, which of two or more land lines as well as cable routes he might desire a message sent by, in addition to his right to designate the coast station through which the message should be sent.

SPECIAL LONG-DISTANCE WORK FOR SHIPS.

Considerable discussion was aroused by a proposition which would permit a ship to communicate with another than the nearest coast station. Nations whose shores are washed by narrow seas desired to have their ships communicate direct with their coast stations, over the heads, so to speak, of intervening stations of another country. It was reported that the Dutch station at Scheveningen would frequently communicate with ships to the westward of the English Channel and with ships in the Mediterranean the ships using the ordinary commercial communicating wave lengths. This was probably always at night. England, France, and other nations interested, declared that this long range communication not only disturbed the working of other ships and stations, but deprived them of a

certain amount of revenue, which should be their due under the rule of the Berlin Convention requiring ships to communicate with the nearest coast station, or to hold the messages to be sent to a particular coast station until that coast station became the nearest. Various propositions were advanced, and most of them hinged on the question of preventing interference with the working of a certain station by a ship using unusually high power in its neighborhood, and it was seen that unanimity on the question was not to be expected on account of the diverse interests involved. The United States offered the first compromise, that is, that both ship and station should carry on long-range work on a wave length greater than 1,600 meters. This would minimize interference with communications on 300 or 600 meters, and the long waves would be more suitable for long-range work, especially by day, on account of the lesser daylight absorption of the energy sent out on long waves. This was accepted and 1,800 meters was assigned for this service. There yet remained the question of the distance of a ship from any coast station within which the ship should not send at power greater than 1 kw.

Germany, Austria, Hungary, Holland, Roumania, Denmark and the United States declared that the distance should not be more than 15 nautical miles; Russia, Italy and Portugal declared for 50 nautical miles; Spain for 150, but later agreed on 50. The matter was advanced by Great Britain proposing a compromise, naming 30 miles. After extended debates, during which it was only by the exercise of considerable patience and appeals to the conciliatory spirit shown throughout the sittings of the conference, the following wording was adopted, following the clauses relating to a ship sending to the nearest shore station:

"Exceptionally, transmission may be made to a more distant coast station, provided—

"(*a*) That the radio-telegram is intended for the country to which such coast station is situated and that it comes from a ship subject to that country;

"(*b*) That for calls and transmission both stations use a wave length of 1,800 meters;

"(*c*) That transmission by this wave length does not disturb any transmission made, by means of the same wave length, by a nearer coast station;

"(*d*) That the ship station is more than 50 nautical miles distant from any coast station shown in the nomenclature. The distance of 50 miles may be reduced to 25 miles, subject to the reservation that the maximum power at the terminals of the generator do not exceed 5 kilowatts and that the ship stations be established in conformity with Articles VII and VIII. This reduction of distance shall not apply in the seas, bays or gulfs of which the shores belong to one country only and of which the opening to the high sea is less than 100 miles wide."

OTHER CHANGES AND ADDITIONS TO THE BERLIN CONVENTION.

Ships and shore stations may use either of the two wave lengths assigned for general public correspondence, but any communication must be made by the use of the same wave length. A ship and a station or two ships may change from 300 to 600, or *vice versa*, by mutual consent.

The International Bureau will publish an official map showing coast stations, ranges, steamship lines, etc.

Arrangements must be made for cutting down the power to obtain several ranges less than the normal range, the shortest about 15 miles.

There will be two classes of certificates for operators: the first class for a speed of 20 words per minute, the second for from 12 to 19 words. The latter may be employed on ships not handling messages for the public, or as a second operator or a substitute on other vessels.

Ships will be placed in three classes:

1. Ships with a constant radio watch;
2. Ships with limited working hours;
3. Ships having no fixed working hours.

Those in the second class must work continuously in the specified hours and must listen for distress calls for the first

10 minutes of each hour of other hours. The time to be used was not stated, and this is an advantage, as the different local times kept may result in an almost constant watch in a given area. Each Government is to specify the class for each of the ships licensed.

The date and hour when a radiogram is handed in on board a ship is to be transmitted in the preamble.

A ship will be allowed to transmit a radiogram to the nearest coast station by relaying through not more than two ships.

Ships or stations intending to use high power shall make a warning signal — — . . — — three times.

The items of information required of a ship calling a station have been simplified.

Certain special radiograms not admitted by the Berlin Convention will be accepted, designated:

1. Reply-paid;
2. Collated;
3. Special delivery;
4. For delivery by post;
5. Multiple;
6. With acknowledgment of receipt;
7. Paid service advices;
8. Urgent.

Communication between ship and ship must be carried out without interfering with the service of coast stations, the latter having, generally, the right of way.

## THE INTERNATIONAL BUREAU.

The International Bureau of the Telegraphic Union was entrusted with the duty of collecting, co-ordinating and publishing information of every kind relating to radio-telegraphy. Its duties are purely administrative.

A committee examined the accounts of the Bureau, and the management was highly commended for its efficient and economical administration. The allowance per annum for its expenditures were increased from 40,000 to 80,000 francs, the United States assuming six shares of the first class for

herself and territories, on account of the importance of radio-telegraphy to their development and defense.

After certain three-letter calls had been assigned to various countries, the United States getting the calls commencing with N and W, the assignment of additional calls required for future needs was delegated to the Bureau.

### RESERVATION REGARDING RATES

In order to simplify the handling of the rate question in this country under the different conditions existing, the American delegation reserved for the Government the right to fix rates independently of those required by the Convention.

### THE CONFERENCE OF 1917.

Just prior to the final adjournment, on July 5, acting under the instructions of the State Department and in accordance with the privately expressed wishes of the various delegations, an invitation was extended to the nations represented to attend a third international conference at Washington. It was first suggested that the date be fixed for 1915, the year of the official opening of the Panama Canal and the Panama-Pacific Exposition, but the conference considered that at least five years should elapse before another conference should be held. The invitation was greeted with loud applause. It was explained that the invitation was tentative and that it would need confirmation by the United States Government after approval by Congress.

It is to be hoped that long and careful preparation will be made for the accommodation and entertainment of foreign delegations. Both at Berlin and at London the place and times of meeting and various conveniences for the delegates were carefully planned. The entertainments were all on a very generous scale, and a liberal appropriation will be needed to approach the degree of hospitality extended by the British Government. There were three dinners with an average attendance of about 300 persons, all on the " ban-

quet" order. Many excursions to points of interest in the vicinity of London, and visits to places of amusements were carefully arranged. In addition, the commercial radio companies entertained all the delegates by excursions to their plants and by dinners. A three-day excursion to the radio station at Poldhu in Cornwall was given by the Marconi Co. This plant is similar to that near South Wellfleet, on Cape Cod, and is used for sending press dispatches every night to transatlantic steamers, Poldhu covering one-half of the ocean and South Wellfleet the other half.

Above all, what we shall require will be an able linguist, diplomat, and parliamentarian to act as president of the conference, who should be designated long enough in advance to enable him to learn the language of radio-telegraphy; and a number of carefully thought-out propositions backed by energetic men who are familiar with the French language.

# NOTES.

## THE PLACE OF MATHEMATICS IN ENGINEERING PRACTICE.*

By Sir William H. White, K. C. B.

The foundations of modern engineering have been laid on mathematical and physical science; the practice of engineering is now governed by scientific methods applied to the analyses of experience and the results of experimental research. The Charter of the Institution of Civil Engineers defines engineering as the "art of directing the great sources of power in Nature for the use and convenience of man." Obviously such direction can only be accomplished by engineers who possess an adequate acquaintance with "natural knowledge"—with the laws which govern these great sources of power. Obedience to natural laws is a condition essential to the full utilization of the great sources of power. It is true, no doubt, that notable achievements in engineering were accomplished during the last century by men whose education was imperfect, whose mathematical and scientific knowledge was small, whose appeal to past experience gave little assistance in the solution of new problems. Their successors now enjoy greatly superior educational advantages; they can profit by enormous advances made in all departments of science and manufacture; they can study and criticise work done by their predecessors in the light of long subsequent experience; but even now there is room for surprise, if not for wonder, when one realizes the great success attained by these early engineers.

The advantages obtainable by the combination of scientific training with practical experience are, however, in no way depreciated because, through force of circumstances, the pioneers of engineering had to do their work as best they could. Not a few of these men recognized the serious disadvantages resulting from their lack of scientific training and gave valuable assistance to a movement which ultimately led to the existing methods of training. George Stephenson, for example, who grew to full manhood practically uneducated, took care to secure for his son Robert Stephenson the advantages of a good elementary education, which was followed by a period of practical training and then by a course of scientific study at Edinburgh University. The careers of both father and son were greatly influenced by this action, and it is of interest to note that the scheme which Stephenson framed and carried out for the professional training of his son—including the alternation of scholastic and engineering work—was, in its essential features, identical with that recommended in the report of a representative Committee of British Engineers over which I had the honor to preside eight years ago. That report has been approved by the leading engineering institutions of the United Kingdom and is now largely influencing the education of engineers.

The fundamental idea underlying the accepted system of training engineers consists in the combination of an adequate knowledge of the sciences which bear upon engineering with a thorough practical training on actual engineering works. No man is now entitled to admission as a corporate member of the Institution of Civil Engineers unless and until he has given proof of the possession of both these qualifications. Neither kind of training standing alone or when developed disproportionately can be re-

* Lecture delivered at the Mathematical Congress, Cambridge, August 24th, 1912.

garded as satisfactory or as meeting the needs of engineering practice. Formerly undue prominence was given to practical training and experience, while the facilities for scientific training were at first non-existent and for a long period were inadequate. Then came a better appreciation of scientific method and a great development of technical education, departments being established for the teaching of engineering science in universities and university colleges. The utilization of these opportunities for instruction by considerable numbers of young men not unnaturally brought about a swing of the pendulum which went beyond reasonable limits. For a time there was a tendency to exalt unduly scientific education and to depreciate the value of practical training. The hard pressure of experience has done much to adjust this disproportion. University graduates when they enter upon actual work soon discover that degrees in engineering, valuable as they undoubtedly are, require to be supplemented by thorough practical training. On the other hand, men who begin their engineering careers as pupils or assistants to practising engineers or as members of engineering office staffs become convinced that their limit of possible attainment must be low unless scientific knowledge is added to practical experience. Under existing conditions new and difficult problems continually arise in all branches of engineering practice, and satisfactory solutions can only be found by bringing to bear upon these problems all the resources furnished by natural knowledge, accumulated experience, and experimental research.

The full equipment of an engineer must include knowledge of other sciences besides the mathematical, but our present concern is exclusively with the latter. An adequate knowledge of mathematics must be possessed by every educated engineer, because he thus acquires valuable tools, by the use of which he can overcome difficulties which would otherwise be insuperable, as well as habits of thought and methods of rigorous investigation which are invaluable when he has to deal with novel and difficult undertakings. Apart from the employment of mathematics it would not be possible for the engineer to carry out designs and construction of engineering works, of structures and machines, capable of fulfilling their intended purposes and possessing both sufficient strength and durability. The days of blind reliance upon engineering formulæ and "rules of thumb" are over. Syllabuses of instruction for the guidance of engineering students, standards established for degrees and diplomas in engineering science, conditions laid down as necessary qualifications for membership of great engineering societies, all furnish full recognition of the fact that an adequate knowledge of mathematics is essential to the successful practice of engineering.

It may be asked what range and character of mathematical knowledge can be described as adequate. Answers to this question are to be found in calendars of schools of engineering, which set forth detailed courses of study considered necessary for the attainment of degrees or diplomas. Identity of conditions does not exist in these regulations, but a closer approach to uniformity has been reached as greater experience has been gained, and it is obviously desirable that further progress should be made in that direction. Engineering degrees ought to be based on a common standard and to represent an equal attainment. These degrees, of course, should be regarded simply as certificates of knowledge of the fundamentals of engineering science; they do not cover all the mathematical knowledge requisite for the practice of particular branches of engineering, and in most branches a greater range of mathematics is necessary. Moreover, a degree course in engineering requires to be supplemented in all cases by subsequent practical training and experience, and in many cases by advanced or specialized courses of mathematical study going beyond the standards associated with degrees. In the settlement of these advanced courses the needs of each branch of engineering must be determined on

the basis of experience, and the subject is one to be dealt with satisfactorily only on the basis of conference between practising engineers and mathematicians. The former know the needs which must be met, the latter can advise as to the best methods of meeting requirements.

Differences of opinion have always prevailed, and still exist, in regard to the methods by which mathematics should be taught to engineering students. Some authorities favor the arrangement of specialized courses of instruction—"mathematics for engineers" or "practical mathematics"—and advocate the creation of separate mathematical sections for engineering schools, even when these schools form departments of universities or institutions which possess well-organized mathematical departments. Other persons, whose opinions are entitled to equal respect, believe that purely mathematical instruction is best given to engineering students by mathematicians, and that a similar rule should apply to instruction in other branches of science, because that method must lead to a broader view of science and a greater capacity for original and independent investigation than can be obtained by specialized teaching narrowed down to the known requirements of previous engineering practice. Personal experience and observation—as student, teacher, and practising engineer—lead me to rank myself with the supporters of the latter method of teaching mathematics. No doubt, in the actual practice of engineers there is room for "short cuts" and for special methods in the use of mathematics; but I am convinced that during the period of education it is advantageous to follow ordinary methods of teaching and to leave specialization for the time when the performance of actual work will almost inevitably lead each individual to make his choice of the branch of engineering to be followed, and of the methods which will best economize his labor and time in doing the work of calculation. The trend of professional opinion certainly lies in the direction of utilizing as far as possible existing mathematical departments for the instruction of engineers. During the past year the subject has been exhaustively considered by the Governors of the Imperial College of Science and Technology, a special committee having been appointed for that purpose. The Imperial College, as is well known, has been formed by bringing together the Royal College of Science, the Royal School of Mines, and the Engineering College founded and maintained by the City and Guilds of London Institute. The last-mentioned College at the outset had to be necessarily self-contained, and had its own mathematical department, which was admirably organized and conducted by Professor Henrici over a very long period. The Royal College of Science and Royal School of Mines also included a department of mathematics and mechanics, of which Professor Perry has been the distinguished director for many years past. Both these departments have justified their existence and done admirable work, but the development of the scheme of the Imperial College rendered it necessary to reconsider the subject of future mathematical instruction in the College as a whole. Alternatives taken into consideration were: (1) The continuance of separate provision for engineering students; (2) the creation of a separate department to be presided over by a mathematician of distinction, in which engineering students would receive their fundamental instruction in mathematics. After thorough investigation the latter course was preferred and will be carried into effect. Its adoption will in no way interfere with the teaching of special applications of mathematics as parts of the courses of instruction given by professors of engineering, and no one familiar with the training of engineers would consider such a change desirable.

A second example of the opinion which now prevails respecting the teaching of mathematics to engineers may be found in a valuable paper by Professor Hopkinson, of Cambridge University, published this year as one of the series of special reports to the Board of Education on the

teaching of mathematics in the United Kingdom. Professor Hopkinson states that for good reasons and for a considerable period after the engineering department was established at Cambridge its students, with few exceptions, "got the whole of their instruction within the walls of the Engineering Laboratory," and "had not the full advantage of their position as students of engineering science in a center of mathematical learning and research." In recent years, however, "the establishment of closer relations between the two studies—mathematics and engineering—has made great progress, and at the present time the students of engineering get their foundation of mathematics and elementary mechanics from college teachers, many of whom have graduated in the mathematical tripos."

It may be interesting to add that in a recent "Summary Report of the Teaching of Mathematics in Japan," Professor Fujisawa has discussed the "teaching of mathematics in technical education" in a brief but interesting fashion. While advocating the "practical" method of instruction, he is careful to explain that the form of utilitarianism which he recommends "has the potentiality of manifesting its usefulness wherever there is a necessity"—a condition which obviously cannot exist unless the engineer is endowed with a good knowledge of the principles and processes of mathematics.

More than seventy years ago men had received a mathematical and scientific training in the first British School of Naval Architecture, wrote as follows in vindication of the necessity for the liberal education of those engaged in the designing of ships:—"The study of naval architecture brings early conviction to the mind of the constructor that he can trust little or nothing to *a priori* reasoning. He uses the exact sciences, it is true, but uses them only as a means of tracing the connection between cause and effect in order to deduce principles that may be applied to his future work, with a certainty of producing the results he contemplates." This utterance may be applied with equal force to the practice of other branches of engineering, and amongst the exact sciences which play so important a part in successful achievement mathematics certainly hold the first place. The "complete engineer" of the earlier and simpler periods can exist no longer under modern conditions. Even the ablest men are driven to specialize in practice. But whatever branch of engineering may be selected the worker will need that fundamental training in mathematics to which allusion has been made. Few engineers engaged in professional work have opportunities of prosecuting mathematical studies systematically, although they are continually using mathematical tools provided during their college careers, and not infrequently have occasion to add to their mathematical equipment in order to meet new demands, or to go beyond precedent and experience. When one considers the great responsibilities which practising engineers have to bear, it is not surprising to find that they, as a class, have made comparatively few contributions to the advancement of mathematical science, although they have been well trained in mathematics and continually apply that knowledge. There are, of course, exceptions to this rule; indeed, I have known engineers who turned to mathematics as a recreation, but these men are exceptional. Another group, whose members have done notable work of a mathematical nature, have been trained as engineers, but have either passed out of practice to an extent which left them ample leisure or have become professors of engineering. The names and work of engineers like Rankine, William Froude and John Hopkinson will always be held in honor by mathematicians as well as by members of the profession which they adorned. The labors of many mathematicians who have devoted themselves to the tuition of engineers, and after becoming acquainted with the problems of engineering, have done splendid work in the foundation of mathematical theories, on which they have based valuable rules of a practical nature, also deserve and will always receive

the grateful appreciation of engineers. But, speaking broadly, there is a real and abiding distinction between the engineer, however accomplished he may be in mathematical science, and the mathematician, however well informed he may be in regard to engineering practice. The mathematician necessarily regards engineering chiefly from the scientific point of view, and although he may aim at advancing engineering science, he is primarily concerned with the bearing of mathematics thereon. The engineer, being charged with actual design and construction of efficient and permanent works, in the most economical manner possible, must put consideration of a practical and utilitarian character in the forefront; and while he seeks to utilize the aids which mathematical and physical science can render, his chief aim must always be to achieve practical and commercial success. There is obviously room for both classes, and their close and friendly collaboration in modern times has produced wonderful results. Fortunately, the conditions which formerly prevailed have ceased to exist. Much less is said about the alleged distinction between pure and applied science, or about the comparative untrustworthiness of theory as compared with practical experience. Fuller knowledge has led to a better understanding of what is needed to secure complete success in carrying out great engineering works, on which the comfort, safety, economical transport and easy communications of the civilized world so largely depend. The true place of mathematics in engineering practice is now better understood; and it is recognized to be an important place, although not so important as was formerly claimed for it by mathematicians.

The character of the change which has taken place in the use of mathematics in connection with engineering practice may be illustrated by reference to that branch of engineering with which my life work has been connected. It is probably true to say that no branch of engineering has benefited more from mathematical assistance than naval architecture has done, and naval architects undoubtedly require to have at least as intimate a knowledge of mathematics as any other class of engineers. Moreover, it was one of the first branches of engineering for which the foundation of a mathematical theory was attempted in modern times; and these attempts were made by men who in their day and generation were recognized to be in the first rank of mathematicians. The work which they did is now almost forgotten, but it laid the foundation for the science of naval architecture as it exists today. To France belongs the honor of having given most encouragement to men of science to attack these problems, and the Academy of Sciences aided the movement greatly by offering prizes which brought into the field not a few of the ablest European mathematicians during the latter half of the eighteenth century. Few of these mathematicians had personal knowledge of the sea or ships, and their investigations were influenced by these limitations. Others had made long voyages; like Bouguer, who, in 1735, proceeded to Peru *pour la mesure de la Terre*, and as a consequence of the experience a very practical tone was given to his famous *Traite du Navire*, which was published in 1745. It would be interesting to sketch the valuable work done by this single mathematician, but time does not allow me to do so. My main purpose at present is to illustrate the change that has since taken place in the use of mathematics in attacking engineering problems, and this may be done better by taking a particular problem and showing how it was dealt with in the eighteenth and nineteenth centuries.

Daniel Bernoulli in 1757 won the prize offered for the second time by the Academie Royale des Sciences for an answer to the question: What is the best means of diminishing the rolling and pitching of a ship without thereby sensibly depriving her of any of the good qualities which she should possess? His memoir was published subsequently; it is an admirable piece of work, and deals thoroughly with the stability of ships; but here Bernoulli had been anticipated by Bouguer and made

acknowledgment of the fact. Greater originality was shown in a mathematical investigation of the influence of wave motion upon the conditions of fluid pressure, as well as the determination of the instantaneous position of equilibrium for a ship floating amongst waves. Bernoulli recognized that the particles of water in a wave must be subjected to horizontal as well as vertical accelerations, although in his mathematical expressions he took account of the latter only. He emphasized the important influence which the relative sizes of waves and ships must have upon rolling and pitching motions, and advised that attention should be mainly devoted to cases where ships were small in proportion to the waves they encountered. In this particular he departed from assumptions usually made by his contemporaries and anticipated modern views. Bernoulli also dwelt upon the critical case where ocean waves forming a regular series have a period synchronizing with the period of still-water rolling of the ship which they meet. A gradual accumulation of angular motion was shown to be inevitable in such circumstances, and it was remarked that the consequent rolling motions must be considerable and might possibly become dangerous in their extent. Bernoulli recommended the conduct of experiments to determine the periods of oscillations of ships in still water, and described methods of conducting these experiments. He also insisted upon the necessity for making accurate observations of the rolling of ships when amongst waves, and made other suggestions of much practical value which have since been repeated by writers unfamiliar with Bernoulli's work and have been practically applied. Unfortunately, the neglect by Bernoulli in his mathematical investigations of the horizontal accelerations of particles of water which he recognized as existing in waves, led to erroneous conclusions in regard to the instantaneous position of equilibrium for a ship when floating amongst large waves and the best means for securing steadiness. Bernoulli considered that when a ship was in instantaneous equilibrium her center of gravity and the center of buoyancy—*i. e.*, the center of gravity of the volume of water instantaneously displaced by the ship—must lie in the same vertical line. This condition, of course, holds good for a ship floating at rest in still water, but not for a ship floating amongst waves of large relative dimensions. Bernoulli deduced from his investigations a practical rule for the guidance of naval architects, viz: that, in order to minimize rolling, ships should be designed so that their centers of buoyancy when they were upright and at rest should be made to coincide with the center of gravity. He considered that ships should be made deep, that large quantities of ballast should be used, and that the cross sections should be approximately triangular in form. This practical rule was misleading, and if applied in a design might be exceedingly mischievous in its effect on the behavior of ships. Bernoulli himself foresaw that in certain cases this rule would work badly, but he considered that these would but rarely occur. It is now known that this view was mistaken.

The detailed mathematical investigations contained in Bernoulli's memoir are still of much interest; they include the examination of cases in which were assumed widely differing ratios of the natural periods of ships to the period of the waves producing rolling motion. Throughout, the motions of ships were supposed to be unaffected by the resistance of the surrounding water, but Bernoulli did not overlook the steadying effect which water resistance would exercise on a ship in a seaway; on the contrary, he recognized the influence which changes in the under-water forms of ships must have upon the amount and steadying effect of that resistance, and he recommended the use of side keels in order to minimize rolling. Having regard to the state of knowledge at the time this memoir appeared it was undoubtedly a remarkable piece of work and it well deserved the reward bestowed by the Academy. It contained many

practical suggestions for experimental inquiry and for guidance in the preparation of designs for ships, but it was essentially a mathematical study and had little influence on the work of naval architects.

A century later the same problems were attacked by William Froude, a graduate of Oxford University, and an engineer of experience in constructional work. As an assistant to Isambard Brunel, the attention of Froude had been directed to these subjects in connection with the design and construction of the *Great Eastern*, a ship of relatively enormous dimensions as compared with contemporary ships and of novel type, respecting whose safety, manageability, and behavior in heavy seas, serious doubts had been expressed. Like Bernoulli, of whose work I feel confident Froude had no knowledge, the modern investigator perceived that amongst waves there must be considerable variations in the direction and magnitude of the pressure delivered by the surrounding water on the surface of a ship's bottom, and that the instantaneous position of equilibrium for a ship exposed to the action of waves of large relative dimensions must be discovered if a theory of rolling was to be framed. Froude worked out a complete theory of trochoidal wave motion and enunciated the principle of an "effective wave slope." In his investigations it was assumed that the resultant water pressure on the ship at each instant acted through the center of buoyancy and normally to the effective wave slope. In the differential equation framed for unresisted rolling Froude took a curve of sines for the effective wave slope instead of a trochoid. Having obtained the general solution of that equation he proceeded to consider the behavior of ships as influenced by variations in the ratios of their still-water periods of rolling oscillation to the relative periods of the waves encountered. In this manner the particular cases considered by Bernoulli were readily investigated and many of the broad deductions made a century before were amended. The critical case of synchronism of ship period and wave period which Bernoulli had brought into prominence was shown to be that requiring most consideration. For that case the increment of oscillation produced by the passage of each wave of a regular series was determined on the hypothesis of unresisted rolling, and was shown to be about three times as great as the maximum inclination to the horizontal of the effective wave slope. It was also made clear that apart from the influence of water resistance such synchronism of periods must lead to a ship being capsized by the passage of comparatively few waves.

Up to this point the investigation made by Froude was strictly mathematical, and the modern engineer who had received a thorough mathematical training had reached results superior to those obtained by the famous mathematician a century before, becoming, in fact, the founder of a theory for the oscillations of ships amongst waves which has been universally accepted. Like Bernoulli, Froude became impressed with the necessity for experiments which would determine the periods of still-water rolling for ships, and with the desirability for making observations of the rolling of ships in a seaway. In addition he emphasized the necessity for more extensive observations on the dimensions and periods of sea waves, a subject which had been investigated to some extent by Dr. Scoresby and other observers, but had been left in an incomplete state. One great generalization made by Froude at an early period in this important work has since become a fundamental rule in the practice of naval architects, viz: that freedom from heavy rolling under the conditions usually met with at sea was likely to be favored by making the period of still-water rolling of ships as large as was possible under the conditions governing the designs. This rule was the exact converse of that laid down by Bernoulli, as the effect of the latter rule by increasing the stability would have lessened the period. The explanation of this simple rule is to be found in the consideration that the longer the natural period

of a ship is the less likely is she to encounter waves whose period will synchronize with her own.

Purely mathematical treatment of the subject did not, however, satisfy the mind of a trained engineer like Froude. For practical purposes it was essential that the effect of water resistance to rolling should be determined and brought into account. Here, purely mathematical investigation could not possibly provide solutions; experimental research, conducted in accordance with scientific methods, became necessary. Aided by the Admiralty, Froude embarked upon a series of experiments which extended over several years. Most of these experiments were made on actual ships, but models were employed in special cases. In the analyses of experimental results mathematics necessarily played a great part. Indeed, without their employment the proper deductions could not have been made. On the basis of these analyses, Froude obtained valuable data and determined experimentally "coefficients" of resistance to rolling experienced respectively by the flat and curved portions of the immersed surfaces of ships. Furthermore, he demonstrated the fact that the surface disturbance produced by the rolling of ships in still water accounted for a large part of the extinctive effect which was produced when a ship which had been set rolling in still water was allowed to come to rest. In this way, and step by step, Froude devised methods by means of which naval architects can now calculate with close approximation the extinctive effect of water resistance for a new design. Finally, Froude produced a method of "graphic integration," the application of which, in association with the calculation of the effect of water resistance, enables a graphic record to be constructed, showing the probable behavior of a ship when exposed to the action of successive waves, not merely when they form a regular series, but when they are parts of a regular sea. Subsequent investigators have devised amendments or extensions of Froude's methods, but in all essentials they stand today as he left them—a monument of his conspicuous ability and an illustration of the modern method in which mathematics and experimental research are associated in the solution of engineering problems which would otherwise remain unsolved.

In tracing, as has been done, the contrast between the methods of Bernoulli and Froude, an indirect answer has been given to the question, "What is the true place of mathematics in engineering practice?" It has been shown that even in the hands of a great mathematician purely mathematical investigation cannot suffice, and that Bernoulli became convinced, in the course of his study of the behavior of ships in a seaway, that no complete or trustworthy solution could be found apart from experimental research, as well as careful observations of ocean waves and the rolling of actual ships. Bernoulli was not in a position to undertake or to lead others to undertake these experiments and observations. In his mathematical investigations he made, and necessarily made, certain assumptions which are now known to have been incorrect. Even the most accurate methematical processes when applied to equations which were framed on imperfect or incorrect assumptions could not produce trustworthy results; and consequently the main deductions made by Bernoulli and the rules recommended by him for the guidance of naval architects would have led to disappointment if they had been applied in practice. On the other hand, Froude, himself a great experimentalist, was fortunately able to impress upon the British Admiralty, through the constructive staff, the importance of making experiments and extensive observations of wave phenomena and the behavior of ships. Not merely did Froude devote many years of personal attention to these subjects, but he was aided over a long period by the large resources of the Royal Navy. Similar work on a very large scale was also done simultaneously by the French Navy. Some of my earliest experiences at the Admiralty forty-five years ago were gained in connection with these observations and

experiments, so that I speak from personal knowledge of the influence which Froude exercised, the inspiration of his great devotion and wonderful initiative. As a result of all these efforts a great mass of experimental data was accumulated; the results of a large number of observations were summarized and analyzed; and, in the end, the soundness of the modern theory was established, and the future practice of naval architects was made more certain in their attempts to produce designs for ships which should be steady and well behaved at sea.

At the risk of making this lecture appear to be chiefly a notice of the work done by William Froude, or a summary of the advances made in the science of naval architecture, another illustration will be given of the general principle laid down in regard to the place of mathematics in engineering practice.

Mathematicians, from an early date, were attracted by the subject of the resistance offered by water to the motion of ships, and made many attempts to frame satisfactory theories. The earliest of these attempts were based upon the assumption that the immersed surface of a ship's skin could be treated as if it consisted of an aggregation of elements, each of which was a very small plane area, set at a known angle of obliquity to the direction of motion through the water. For each elementary plane area it was proposed to estimate the resistance independently of the others, and as if it were a small isolated flat plate. The integration of such resistances over the whole surface was supposed to represent the total resistance of the water to the motion of the ship at a given speed. Certain further assumptions were made in regard to the laws connecting the resistance of each unit of area with its angle of obliquity to the direction of motion and with the speed of advance. The effect of friction was, in most cases, neglected, nor was any account taken of surface disturbance produced by the motion of the ship. It is unnecessary to dwell upon the errors and incompleteness of these assumptions. So long as ships were propelled by sails little practical importance attached to an exact determination of the resistance experienced at a certain speed. When steamships came into use it was of primary importance to have the power of making close approximations to that resistance, because estimates for the engine power required to attain a given speed had to be based thereon. The subject received great consideration, as the result of which certain simple rules were framed and commonly employed in making estimates for the engine power to be provided in new ships. These rules were mainly based on the results obtained by trials of existing vessels, and these trial results, of course, included not merely the effect of water resistance—as influenced by the form and condition of the immersed surface of a ship—but were also affected by the varying efficiency of the propelling apparatus and propellers. Many attempts were made to separate these items of performance and to determine the actual amount of the resistance for a ship and the separate efficiencies of her propellers and machinery. Little progress was made until 1868. Mathematical theories were framed, it is true, for estimating the efficiency of propellers, but while these theories were accurate enough if the assumptions underlying them had been complete and representative of actual phenomena, there was no possibility of fulfilling those conditions, since the phenomena were neither fully known nor understood.

In 1868 a special and representative committee—including Rankine and Kelvin—appointed by the British Association, made a report on this subject and recommended that towing experiments should be made on full-sized ships. The committee was almost unanimously of opinion that the only method which would give trustworthy information in regard to the resistances experienced at various speeds was to tow actual ships and not to depend upon models. William Froude dissented from this conclusion and recommended model experiments. Accepting the stream-line

theory of resistance which Rankine had introduced, Froude based upon it a system of experiment, which dealt separately with frictional resistance and applied to the residual resistance—after friction had been allowed for and deducted—a law of "corresponding speeds" between models and full-sized ships which he had worked out independently. That law had been previously recognized in a more general form by mathematicians, and had been investigated for this particular case by a French mathematician, M. Reech, of whose work Froude was then ignorant. By this happy association of mathematical theory with experimental research Froude placed in the hands of naval architects the means of solving problems which could not be dealt with either by purely mathematical investigation or by experience with actual ships. Experimental tanks of the character devised by Froude have now been multiplied in all maritime countries. The latest and in many respects the best of these tanks, which is due to the generosity of Mr. Alfred Yarrow, is a Department of the British National Physical Laboratory. The operations of these tanks have resulted in a great addition to natural knowledge and have secured enormous economies of fuel. The success achieved in connection with modern developments of steam navigation and the attainment of very high speeds is chiefly due to tank experiments which have involved very small cost, and enabled naval architects to choose for every design the form which gives the least resistance possible under the conditions laid down for a new ship, even when the size or speed required go beyond all precedent. Considerations of stability, carrying capacity, available depths of water, dimensions of dock entrances and other matters, as well as speed and fuel consumption, may limit the designer and narrow the alternatives at his disposal; but ordinarily there is room for considerable variations of form in a new design, and in making the final selection of form it is essential that the designer should know how the resistances of these permissible alternatives compare. Naval architects throughout the world enjoy great advantages in this respect over their predecessors, and owe their position entirely to the genius and persistence of William Froude.

Since the work of Froude in this direction was done model experiments have become the rule in many departments of engineering, and the scientific interpretation of the results has greatly influenced the designs of structures and machines. Prominent amongst these recent applications of experimental research on models stand those relating to air resistance and wind pressure on bridges and other structures. In regard to the laws of wind pressure much has been done in recent years, and in connection with the effects of wind pressure on engineering structures especial reference ought to be made to the work done by Dr. Stanton at the National Physical Laboratory. All engineers owe a debt of gratitude to that distinguished experimentalist and to the Institution where he works, and they recognize the fact that he has demonstrated the trustworthiness of deductions made from tests with small models exposed to the action of air currents when applied on the full scale to complicated structures for which independent mathematical calculations of the effect of wind pressure could not be made. The late Sir Benjamin Baker, who was chiefly responsible for the design of the Forth Bridge, was one of the first to appreciate and make use of this experimental system, and no engineer of his time more frankly admitted than he did what a debt engineering practice owed to mathematics when used in the proper manner.

The proper use of mathematics in engineering is now generally admitted to include the following steps. First comes the development of a mathematical theory, based on assumptions which are thought to represent and embody known conditions disclosed by past practice and observation. From these theoretical investigations there originate valuable suggestions for experimental inquiries or for careful and extensive ob-

servation. The results obtained by experimental research or from observation and experience must be subjected to mathematical analysis, and the deductions made therefrom usually lead to amendments or extensions of the original theory and to the device of useful rules for guidance in practice. Purely mathematical theories have served, and still serve, a useful purpose in engineering; but it is now universally agreed that the chief services of mathematics to engineering are rendered in framing schemes for experimental research, in analyzing results, in directing the conduct of observations on the behavior of existing engineering works, and in the establishment of general principles and practical rules which engineers can utilize in their daily professional employment.

One of the most recent examples of this procedure is to be found in the constitution and proceedings of the Advisory Committee appointed in 1909 by the British Government in connection with the study of aeronautics. Its membership includes distinguished mathematicians, physicists, engineers and officers of the Army and Navy, and its president is Lord Rayleigh, Chancellor of the University of Cambridge. The declared intention in establishing this committee was to bring the highest scientific talent "to bear on the problems which have to be solved" in order to endow the military and naval forces of the British Empire with efficient aerial machines. The reports published during the last two years are of great value; the work done by the Committee—described as "the scientific study of the problems of flight with a view to their practical solution"—has been accompanied and supplemented by research and experiment carried out by the Director of the National Physical Laboratory (Dr. Glazebrook) and his staff in accordance with a definite program approved by the Committee. These investigations necessarily cover a very wide field in which there is ample room for the operations of all the branches of science and engineering represented on the Committee, and there can be no doubt that already the influence of the work done has been felt in practice. No one who has followed the progress made in aerial navigation, however, can fail to be convinced that although a considerable amount of purely mathematical investigation has been devoted to the problems of flight, it has hitherto had but little influence on practice in comparison with that exercised by improvements due to mechanical engineering—tending to greater lightness of the engines in relation to their power—and by actual experiments made with models and full-sized flying machines. A stage has been reached, no doubt, where the interpretation by mathematicians of the experimental results available and their suggestions as to the direction in which fuller experimental research can best be carried out are of great importance, and that fact is universally recognized by engineers.

Even when the fullest use has been made of mathematical science applied in the best way, and of experimental research, there still remain problems which have hitherto defied all efforts at their complete solution, and engineers have to be content with provisional hypotheses. Of the James Forrest Lectures given annually at the Institution of Civil Engineers, a long series has been devoted to the description of "Unsolved Problems in Engineering." Mathematicians seeking fresh fields to conquer might profitably study these utterances of practising engineers of repute. On this occasion it must suffice to mention two subjects on which additional light is still needed, although they are now less obscure than they have been in the past, thanks to long years of work and experiment. The first group has relation to the laws which govern the efficiency of screw propellers when applied to steamships, and has long engaged the attention of a multitude of writers in all maritime countries. Many mathematical theories have been published, which are of interest and value as mathematics, and are sound if the fundamental assumptions made could be accepted. It is, however, no exaggeration to say that at

the present time there exists no mathematical theory which has any considerable influence on the design of screw propellers and the determination of the form, area and pitch. Experience and experiment are still mainly depended upon when work of that kind has to be undertaken. Of course certain mathematical principles underlie all propeller designs, but the phenomena attending the operation of a screw propeller at the stern of a ship in motion are too variable and complex to be represented by any mathematical equation, even if they were fully known and understood—which they are not. The water in which a propeller works has already been set in motion by the ship before it reaches the propeller, and the "wake" of a ship in motion is in a very confused state. The action of a propeller upon the water "passing through it" and the manner in which its effective thrust is obtained still remain subjects for discussion and for wide differences of opinion between mathematicians and experimentalists who have seriously studied them. Froude initiated a system of model experiments for propellers both when working in open water and when attached to and propelling ships or developing an equivalent thrust. His son, Mr. R. E. Froude, has done remarkable work in the same direction, and many other experimentalists have engaged in the task; but after more than seventy years of experience in the practical use of screw propellers we remain without complete or accurate knowledge which would enable the designs of propellers for new ships of novel types or of very high speed, to be prepared with a certainty of success. On the whole, naval architects and marine engineers depend largely upon the results of experience with other ships. Although model experiments are also utilized, there is not the same confidence in passing from results obtained with model propellers to full-sized ships. Probably this fact is chiefly due to the essential differences in the reactions between the water and the surfaces of models and the surfaces of full-sized screws moving at the rates of revolution appropriate to each. These matters are receiving, and have already received, careful study, not merely by the superintendents of experimental tanks, but by practising engineers like Sir Charles Parsons and Sir John Thornycroft. The phenomenon of "cavitation," which has been described as the breaking away of water from the screw surfaces when the rate of revolution of the screw exceeds certain limits, and when the thrust per unit of area on the screw exceeds certain values, is one which has given much trouble in the cases of vessels of exceedingly high speed, such as destroyers and swift cruisers. It is being investigated experimentally, but up to the present time no general solution has been found. In existing conditions surprising differences in the efficiency of propellers have been produced by what appeared to be small changes in design. On the whole, the largest improvements have been obtained as the result of full-scale trials made in ships, although model experiments have been of service in suggesting the direction in which improvements were probable. In my own experience very remarkable cases have occurred, and not infrequently it has been difficult, even after the event, to explain the results obtained. One such case may be mentioned as an illustration. A large cruiser obtained the guaranteed speed of 23 knots on trial with a development of about 30,000 horsepower. I had anticipated a higher speed. Progressive trials made at various speeds showed that the "slip" of the propellers became excessive as the maximum speed was approached, although the blade area given to the propellers on the basis of past experience was adequate for the power and thrust. The blade area was increased about 20 per cent., the diameter and pitch of the screw propellers being but little changed. With these new propellers the maximum speed became 24 knots, and 23 knots was obtained with a development of 27,000 horsepower, as against 30,000 horsepower required with the original screws. The increase of blade area necessarily involved greater frictional losses

on the screws, yet the effective thrust was increased, and not only was a higher maximum speed attained, but the power required at all speeds down to 15 knots, became less than in the earlier trials. This incident could be paralleled from the experience of many naval architects, and it illustrates the uncertainties which still have to be faced in steamship design when unprecedented speeds have to be guaranteed.

This open confession of lack of complete knowledge in the presence of the professors of an exact science such as mathematics may be thought singular. It is the fashion to criticise, if not to condemn, designers of ships and their propelling apparatus on the ground that after long experience there ought to be a complete mastery of these problems. That criticism, however, is hardly fair, because it overlooks the fact that throughout the period of steam navigation there has been incessant change in the dimensions, forms and speeds of ships and in the character of the propelling apparatus.

Knowledge is also incomplete, and possibly complete knowledge will never be obtained, in regard to the stresses experienced by the structures of ships at sea, when driven through waves and made to perform rolling, pitching and heaving movements simultaneously. The subject has long engaged the attention of mathematicians and naval architects. Early in the last century Dr. Young made a study of the causes of longitudinal bending in wood-built ships, and presented a memoir to the Royal Society. The eminent French geometrician, Charles Dupin, also dealt with the subject which had great practical interest at a date when serious "hogging" or "arching" of ships was a common occurrence. Since iron and steel have been available for ship construction—and as a consequence the dimensions, speeds and carrying powers of ships have been enormously increased—questions of a similar character have arisen on a larger scale, and have been carefully studied. There is much in common between ship construction and bridge construction under modern conditions; and because of this resemblance engineers practising in works of a constructional nature on land have been brought into close relation with the structural arrangements of ships. Sir William Fairbairn, who was associated with the younger Stephenson in the construction of the Menai and Conway tubular bridges, and Isambard Brunel, whose chief work was on railways, but who designed the famous steamships *Great Western*, *Great Britain* and *Great Eastern*, are amongst the men of this class who have most influenced shipbuilding. There are, however, obvious and fundamental differences between the conditions of even the greatest bridge founded on the solid earth and those holding good in the case of self-contained floating structures carrying great loads across the sea, containing powerful propelling apparatus, and necessarily exposed to the action of winds and waves. In the former case bending moments and shearing stresses which must be provided against can be closely estimated, and ample strength can be secured by adopting proper "factors of safety." In the case of ships no similar approximations are possible, because their structures are stressed not only by the unequal distribution of weight and buoyancy, but have to bear rapidly-varying and compound stresses produced by rolling and pitching motions, by external water pressure and by the action of the propelling apparatus, as well as to resist heavy blows of the sea. Inevitably, therefore, the naval architect has to face the unknown when deciding on the "scantlings" of various parts of the structure of a new ship, which is novel and goes beyond precedent.

Mathematicians have had the courage to attack these problems and to propound theories respecting them. Professor Kreloff, of the Imperial University, St. Petersburg, has been one of the latest workers in this field and has probably carried the mathematical theory furthest; but his work, like that of his predecessors, has had little effect on the practice of naval architects. Indeed, it seems too much to expect that even the most accom-

plished mathematician can deal satisfactorily with the complex conditions which influence the variable stresses acting, from moment to moment, on the structure of a ship at sea. In these circumstances naval architects have been compelled to fall back upon experience with ships which have been long in service at sea, and to obtain the best guidance possible from the application of mathematics to the analysis of that experience and to the device of rules of a comparative nature. In general, this procedure has led to the construction of ships which have possessed ample strength, although the actual margin of strength in excess of the permissible minimum has not been ascertained. In the comparatively few cases wherein weaknesses have been brought to light on service, scientific analysis has enabled even more valuable lessons to be deduced. But it cannot be said that purely mathematical investigation has been of great service to this branch of engineering.

Rankine, many years ago, proposed to base comparison of the longitudinal bending moments and shearing stresses of ships amongst waves on the hypothesis that the distribution of weight and buoyancy should be determined for two extreme cases:—First, when a ship was momentarily resting in equilibrium on the crest of a wave having a length equal to her own length, and a height—hollow to crest—as great as was likely to occur in a seaway, say, one-twentieth of the length of the wave. Second, when she was momentarily floating astride a hollow of such waves. It was recognized, of course, that these cases were purely hypothetical, but the hypothesis has proved of great value in practice. Attempts have been made to carry the calculations further and to take account of the effects of rolling, pitching and heaving motions, and of variations in the direction and amount of water pressure consequent on wave motion. Practice has been influenced but little by these attempts, which have necessarily been based on more or less arbitrary assumptions themselves not free from doubt. On the other hand, Rankine's method has been widely used by naval architects, and the accumulated results of calculation obtained for ships whose reputations for strength and seaworthiness was good are now available for reference. For new designs calculations of a similar character are made, and by comparison of results with those obtained for completed ships most closely approaching the new design in type and dimensions, the principal scantlings are determined for various parts of the structure. In calculating the strength of ships, they are usually treated as hollow girders exposed to the action of forces tending to cause longitudinal bending. The bending moments and shearing stresses calculated for the two extreme hypothetical conditions above described are used in order to estimate the maximum stresses corresponding thereto in any members of the new ship's structure. A comparison of these maximum stresses (per unit of area of material) with the corresponding figures for successful ships, is taken as a guide for determining the sufficiency or insufficiency of the scantlings proposed for the new vessel. In providing for adequate transverse strength and for margins of strength to meet local requirements, naval architects make separate calculations, but in these cases also are guided chiefly by comparisons based on actual experience with other ships. Mathematicians may regard this procedure as unsatisfactory, but they may be assured that any suggestions for improved or more exact methods which may be made will be welcomed by naval architects provided they command confidence and are capable of practical application.

In considering the relation of mathematics to engineering practice one important fact should always be borne in mind. The mission of engineers as a class is to produce results, "to do things," which shall be of practical service to humanity, and shall ensure safety of life and property. Complete solutions of problems, in the mathematical sense, are not usually to be found by engineers. They have to be content in many cases

with partial solutions and fairly close approximations. It may be taken for granted that engineers desire to perform efficiently the duties laid upon them, and that they are ready to avail themselves of all assistance which can be rendered by contemporary science, and by mathematics in particular. On their behalf it has been my endeavor on this occasion to make suitable acknowledgment of the debt which engineering already owes to mathematics, and to indicate a few of the many problems in which further assistance is needed. All members of the engineering profession will endorse my expression of the hope that the close and friendly relations which have long existed between mathematicians and engineers, and which have yielded excellent results during the past century, will always continue and in future be productive of even greater benefits.—"The Engineer."

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## THE "IDLERS."

MARINE BRITANNIQUE. — *On se fait un plaisir de l'autre côté de la Manche de blaguer les plumitifs « the idlers », chargés des transmissions d'ordres.* (D'après *the Fleet*).

—From "La Vie Maritime."

## OLD BOILERS.

The subject of a proper retiring age for old steam boilers is one which comes frequently to light, and which has been in the past a most fruitful source of controversy. Of course there can be no question as to the propriety of condemning to forced retirement those boilers whose diseases of one sort or another have reached the chronic stage, and are no longer curable, but there is at once the basis for a deal of argument when an inspector approaches the owner of a boiler with the statement that it must be replaced because of old age, especially if it is known to have all the apparent qualifications except youth, for many additional years of service.

In the past many curious properties have been attributed to old boilers. One of the most interesting was the notion that they could not explode violently. It was supposed that an old boiler would merely rupture, allowing the pressure to be relieved much as if the safety valve had opened. This idea was discussed at some length in the "Locomotive" in 1881. It was definitely disproved along with many other fallacies, and much popular mystery concerning boiler explosions, by a series of experiments conducted by Mr. Francis B. Stevens, of Hoboken, the founder of Stevens Institute, and Prof. R. H. Thurston, at the Sandy Hook proving grounds, in the fall of 1871. These tests consisted of a series of prearranged boiler explosions in which old boilers, and some new boiler elements, were exploded by raising a steam pressure in them sufficient to produce failure. They gave the first conclusive proof of the fact that a boiler filled with water to its normal level could explode, and also that a boiler might explode violently when hot and under steam at a pressure less than that which it had successfully withstood under the ordinary hydrostatic test.

In 1881 Mr. W. B. LeVan, of Philadelphia, proposed to the American Society of Mechanical Engineers* that all steam boilers should be retired at the age of ten years arbitrarily, in much the same way that car wheels and axles are retired after a certain mileage. His reason for this view was his general distrust of boiler inspection, and although he admitted that the use and care a boiler received must influence its life, he was unwilling to believe in the probability of an inspector finding the extent of that influence, and so expected to forestall all danger by his ten-year limit. Of course it was pointed out at that time that such a rule would work great hardship to the owner who used a good boiler well, and would unduly encourage the unscrupulous owner to push his poor boilers, by fair means or foul, to accomplish their utmost in the allotted ten years.

However, there is a border line between the obvious defects an inspector can detect, and that gradual change in the physical character of the metal coming with advanced age and long use, which can only be implied by a knowledge of similar cases. Here it is that an insurance company must at times make a stand for the removal from service of an old boiler, or at least for a great reduction in the pressure at which it is worked. In order to prove the soundness of such rulings old boilers have been tested from time to time, and it is the purpose of the present article to review certain of these tests, and show the character of the evidence upon which these old-age retirements are based.

In general, two sorts of tests can be made. One sort, of which the early tests of Stevens and Thurston are examples, consists in subjecting the entire boiler, considered as an engineering structure, to either a steam, or hydrostatic pressure great enough to cause rupture. The hydrostatic test is usually employed since it permits careful measurements of the strains at various points to be made as the test progresses, and with these an accurate record of the pressures producing them. The other class includes tests of the metal taken from different parts of a boiler, to show its physical and chemical properties, and, if the original condition of the

* "The Lifetime or Age of Steam Boilers," W. B. Le Van, Trans. A. S. M. E. Vol. II, Page 503.

material is known, is of great value. It of course may very well form an addition to a test of the first sort. Within a short time five old boilers, whose entire history is known, have been tested to destruction by the application of hydrostatic pressure. Three of these, the property of the Oliver Iron Mining Co., of Ishpeming, Mich., were tested by Mr. A. M. Gow, their assistant engineer. The other two were presented to the Bureau of Standards for test by Mr. Nicholas Sheldon, treasurer of the Kendall Manufacturing Co., of Providence, R. I. These boilers were tested to destruction at the plant of W. H. Hicks, boiler makers, Providence, R. I., by Mr. James E. Howard, engineer-physicist of the Bureau of Standards, assisted and advised by Mr. F. B. Allen, vice-president of The Hartford Steam Boiler Inspection and Insurance Co. All five of these boilers had been carried on the books of the Hartford, and had been removed from service at their request.

The boilers tested by Mr. Gow, were known in the records of the Hartford by the numbers 301, 302 and 303, and will be designated in this way. They were nearly identical in construction, of the horizontal return-tubular type, 72 inches in diameter, and 15 feet long. The shells were in five courses, and were made of ⅜-inch plate. The heads were ½ inch in thickness. The longitudinal seams were of the double-riveted lap type fastened with ¾-inch rivets, pitched 2 inches apart, and each boiler was fitted with a cast-iron manhole frame on top of the next to the last course, with a clear opening of about 12×16 inches having its greatest diameter girthwise of the boiler. Two 4-inch cast-iron nozzles were also fitted to each boiler, one on the rear, and one on the second course, for the attachment of the safety valves and steam pipes. The blow-off connections were in the rear heads, and had been used for a long time for the introduction of the feed water. Reference to Fig. 1 will make clear the general arrangement of the boilers, and will indicate their only point of difference, namely, that No. 302 contained 112 three-inch tubes, while both No. 301 and No. 303 were provided with 83 four-inch tubes.

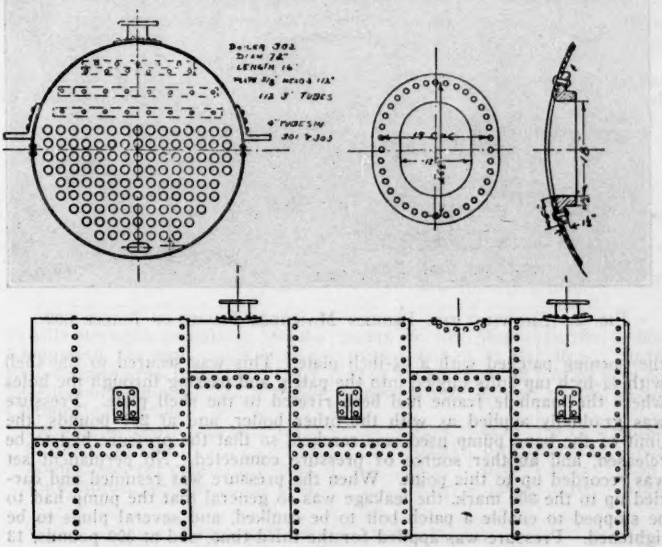

FIG. 1.—DETAILS OF BOILERS 301, 302 AND 303.

The report of the Hartford's inspector shows that these boilers, aside from age, were in apparent good order. There was evidence that no rivets had been replaced, and that the seams had never been chipped and caulked. No fire cracks were visible, and although there was a slight indication of over-heating on the bottom sheets of the rear courses, this was considered trivial. The only repairs known to have been made, were several complete renewals of the tubes.

All three boilers appear to have been made by Kendall and Roberts, of Boston, No. 302 about 1877, and the other two about 1879. The steel plates in No. 302 were branded "Bay State Homo," while those of No. 301 and No. 303 bore the brand "Nashua Iron and Steel Co., Nashua, N. H." "Cast Steel 60,000 lbs.," and an encircled Indian's head. Mr. Gow, in his report of the tests published in "Power," gives it as his opinion that these were among the first boilers to be made in this country of Siemens open-hearth steel.

Boiler No. 302 was tested June 6, 1911. Pressure was applied gradually, and at 275 pounds, the manhole frame failed, tearing the adjacent sheet as shown in Fig. 2. A steel tape stretched around the boiler girthwise, showed a stretch of 3/16 inch in circumference just before the rupture, but on the release of the pressure, no permanent set was found, showing that the elastic limit of the plate had not been reached.

Boiler No. 303 was tested the following day, and in order to find if possible other sources of weakness, the manhole frame was removed, and

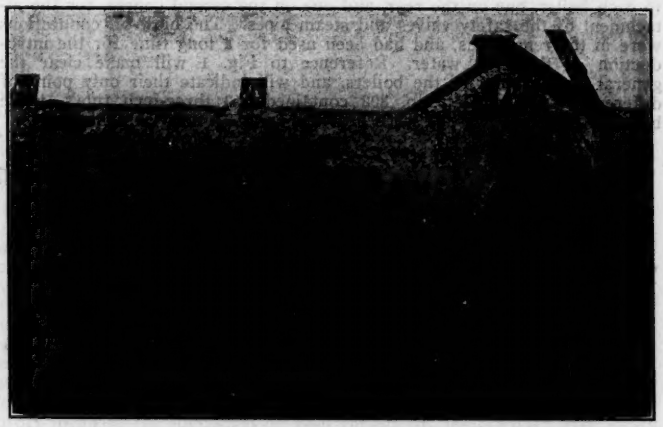

FIG. 2.—SHOWING THE BROKEN MANHOLE FRAME OF BOILER 302.

the opening patched with a ⅝-inch plate. This was secured to the shell with ¾-inch tap bolts tapped into the patch, and passing through the holes where the manhole frame had been riveted to the shell plate. Pressure was gradually applied as with the other boiler, and at 297 pounds, the limit of the hand pump used was reached, so that the pressure had to be released, and another source of pressure connected. No permanent set was recorded up to this point. When the pressure was resumed and carried up to the 300 mark, the leakage was so general that the pump had to be stopped to enable a patch bolt to be caulked, and several plugs to be tightened. Pressure was applied for the third time, and at 300 pounds, 13

patch bolts sheared, the beading at the tube ends started, the tube sheets showed distress, and a permanent set of about 1/16 inch in the circumference of the boiler was recorded. After the release of pressure, the patch was found to overlap the bolt holes about ½ inch. Its appearance after the test is shown in Fig. 3.

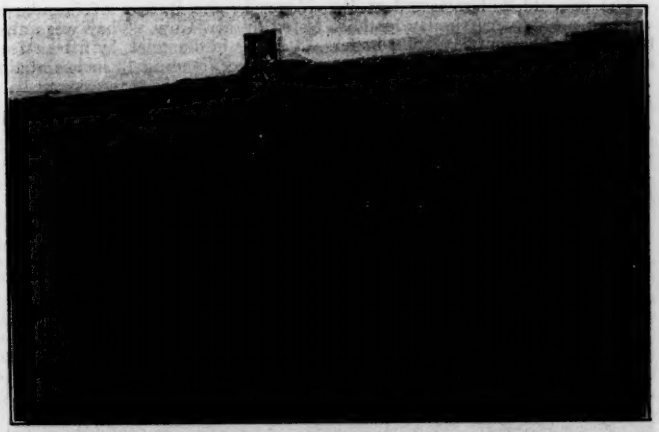

FIG. 3.—APPEARANCE OF BOILER 303 AFTER TESTING.

Boiler No. 301 was tested in its original condition, and failed through the manhole frame at a pressure of 260 pounds. A set of ⅛ inch in the circumference was noted, along with evidences of distress in the longitudinal seams.

Test specimens were cut from these boilers, at points exposed to the action of the fire, and also on the top. These were tested for strength and elongation, and also submitted to a chemical analysis, the results of which are shown in Table I.

TABLE I.

| | Boiler No. 302. | | Boiler No. 303. | | Boiler No. 301. | |
|---|---|---|---|---|---|---|
| | Over fire.* | On top. | Over fire. | On top. | Over fire. | On top. |
| Tensile strength, pounds | 60,460 | 70,145 | 60,186 | 56,400 | 60,780 | 61,680 |
| Elongation, per cent | 22.5 | 20.12 | 21.5 | 27.25 | 26.5 | 19.75 |
| Reduction in area, per cent | 53.7 | 47.05 | 54.52 | 64.88 | 61.62 | 50.80 |
| Elastic limit, pounds | 36,690 | 39,060 | 38,280 | 37,230 | 33,100 | 38,820 |
| Chemical properties: | | | | | | |
| Carbon, per cent | 0.13 | ... | 0.17 | 0.25 | 0.13 | 0.18 |
| Sulphur, per cent | 0.026 | ... | 0.023 | 0.121 | 0.022 | 0.022 |
| Manganese, per cent | 0.27 | ... | 0.29 | 0.37 | 0.20 | 0.28 |
| Phosphorus, per cent | 0.097 | ... | 0.097 | 0.092 | 0.105 | 0.085 |

* Bent cold to 180 degrees without fracture.

These boilers had been designed for a pressure of 100 pounds, but owing to the low factor of safety which they would have at this pressure, due to the low efficiency of the longitudinal joints, they had been worked at a pressure of 80 pounds. At this pressure, the actual factor of safety, based on the pressure of 260 pounds, at which the manhole frame of boiler No. 301 failed, was only 3.25.

The two old boilers tested at Providence by the Bureau of Standards were of a type very similar to those tested by Mr. Gow. They were also five-course, horizontal, tubular boilers, 72 inches in diameter, by fifteen feet long between tube sheets, with the first course extending 12 inches at the

FIG. 4.—BOILER 4,092 PRIOR TO THE TEST. NOTE THE 10-INCH GAGED LENGTHS.

front as a dry sheet. Their appearance prior to the test is shown by Fig. 4. The longitudinal joints were double-riveted lap seams, ¾-inch rivets, pitched 2 inches, placed in punched holes. The plate thickness was ⅜ inch for the shell, and ½ inch in the heads. Both boilers contained 140 3-inch tubes. Domes 2 feet 6 inches in diameter were provided on the second course, and cast-iron manhole frames and safety-valve nozzles were fitted to the middle and rear courses, respectively. Both boilers were made at the shops of the Whittier Machine Co., Boston, Mass., in 1881, of "Benzon" steel. They were known as No. 4,084 and No. 4,092 on the Hardford's records, and were designated by these numbers in the report of the test published in the November, 1911, number of the "Journal of the American Society of Mechanical Engineers."

In this series of tests especial attention was given to measuring the strains and deformations produced in the boiler sheets as the pressure was increased, because in this way a knowledge of the actual behavior of the boiler could be obtained, and so checked up with the strains which might be expected if the ordinary assumptions underlying calculations of boiler strength are justified. To this end small holes were drilled at different points in pairs, exactly ten inches apart, as is shown in Fig. 4. These were then reamed out with a conical reamer, so as to serve for centering two corresponding cone-shaped points on a micrometer strain gage. This in-

strument could be applied to a pair of holes, and their exact distance apart compared to that between an exactly similar pair prepared at the ends of a carefully-measured length in a standard bar. After a stress was applied to the boiler the distance between any pair of holes could again be compared with the standard, and the difference between the two sets of readings would be the stretch. It was said that these measurements were known with a certainty of 0.0001 inch, which is about the stretch which we might expect to find in a bar of steel, 1 square inch in cross section and ten inches long, if it were subjected to a pull of 300 pounds. If we take the stress which will produce a given stretch in a piece of material 1 inch long, and divide this pull or stress by the resultant strain, we get a number known as the "Modulus of Elasticity." In the present case, a stretch of 1/10,000 inch in ten inches, would mean an increase of 1/10 of this, or 1/100,000 inch in a length of 1 inch, and if we were to divide the stress, 300 pounds, by this strain in a 1-inch length, 1/100,000, we should obtain the number 30,000,000 which is the modulus of elasticity for steel. (As a matter of fact the modulus must not be thought of as being obtained from these figures, for of course the pull to produce this stretch of 1/10,000 inch in ten inches was estimated from the modulus obtained by averaging a large number of tests, in which the length of the specimen, the pull applied and the resulting increase of length were carefully recorded.) Knowing the modulus, then, we are in a position to predict the strain which ought to result from any given pressure applied to the inside of a boiler, and if in testing, the actual strains differ from these, we must look for the cause of the rigidity if the strains are too small, or of the yielding if they are too great, and see if the behavior of the metal can be attributed to any peculiarity in the boiler structure which causes a different distribution of stress from that expected. Having outlined the methods of investigation, we will not endeavor to review all the details of the measurements made, but refer the reader who desires to enter into these more fully to the published report, as we are concerned only with the results.

Boiler No. 4,084 was tested first. At a pressure of 266 pounds, leakage along the longitudinal joint of the dome had become so great as to necessitate its removal. The shell was closed with a patch, double riveted, which made use of the same holes as had previously served for fastening the dome. At 270 pounds the cast-iron manhole frame ruptured across the middle of its length and a second patch, closing the opening, was applied in its place. When a pressure of 295 pounds had been reached, 3 front head braces let go and the test was discontinued. The boiler was subsequently dismantled, in order to permit a detailed examination of its interior to be made. Certain regions of distress were revealed through the disturbance of the scale with which the metal was slightly incrusted. This distress was most evident in the dome, at its longitudinal seam, and also under the points of attachment of the lugs, by which the boiler had been supported during the test, and also when in service.

The strain measurements were less comprehensive on this boiler than on No. 4,092, and in general were very similar. One feature was noticed, however, which was absent in the latter case. The gaged lengths, which spanned the longitudinal joints, and therefore measured their slip, decreased with great uniformity from the front towards the rear, suggesting that even though these seams were not directly exposed to the action of the fire, there was a greater range of temperature strain at the front than at the rear. This result is especially interesting in the light of the experiments, reported below, on a French boiler of considerable age, in which it was clearly shown that the deterioration of the metal was closely correlated to its position with respect to the direct action of the fire.

In the hope of attaining higher pressures, boiler No. 4,092 was strengthened prior to the test, by removing the dome and manhole frame, and re-

placing them with patches. The safety-valve nozzle was allowed to remain, but as the test progressed, it was found necessary to replace it with a soft patch, as it was impracticable to caulk the leaks occurring at its junction with the shell. Six 1¼-inch through stays were also added to give additional support to the segments of the heads above the tubes.

Pressure was raised, and at each increment of pressure careful measurements of the various gaged lengths were made, in order to compute the strains. At 300 pounds pressure the safety-valve nozzle had to be replaced, and at 335 pounds the manhole patch failed, shearing its rivets, and tearing the sheet. A section of the sheet was cut out from girth seam to girth seam, and a double-riveted patch inserted in its place. This patch was necessarily hand-riveted, and at the time of publication of the results of these tests higher pressures than 335 pounds had not been attained, due to excessive leakage at this patch.

The results of the strain measurements, of which some 3,300 were taken, may be briefly summarized as follows: The well-known stiffening effect of girth seams and heads were abundantly confirmed, as was the great weakness of the top center-line of the boiler due to the presence of an opening in nearly every course. The double-riveted lap joints were found to give rise to an excessive slip, and the effect of this slip, in producing abnormal stresses in the solid plate abreast the ends of the seam was commented on. It was also evident that since the longitudinal seams in successive courses were only separated by three rivet pitches (6 inches) girthwise, a belt of great tangential weakness existed from end to end of the boiler, and on each side, through these seams.

If a plain cylinder is subjected to an internal pressure the metal ought to contract in length, to make up for its tangential, or round-about extension. Such a contraction occurred in the metal of the boiler shell, but was not uniform; indeed, in the top part of the boiler there was an actual extension. It is also obvious, that if a plain cylinder, like a boiler tube, is subjected to an external pressure, the tube should extend in length, to make up for the girthwise contraction. In the boiler tested such an extension of the tubes was found, though it was modified to some extent by the position of the tube in the shell. Those tubes situated in the center of the nest were in every case extended more than those near the shell, as if the flanged head exerted a restraining influence. It was pointed out, as a matter of fact, that this extension of the tubes, coupled as it was with a contraction lengthwise of the shell, imposed a considerable bending moment on the flanges of the heads.

Let us now, before attempting to form an opinion, or draw conclusions as to the results of these tests, pass on to a consideration of a series of tests of the second sort, made with great care, in which samples of the material of some very old boilers of known antecedents were tested both physically and chemically. It is a point worthy of note that in these tests especial care was taken to keep track of the part of the boiler from which the test specimens were taken, in order that any peculiarity due to exposure either to extreme temperature conditions or to unusual structural stresses might be observed.

These tests, made by Messrs. A. Olry and P. Bonnet, form the subject of an extended report to the (French) Association of Owners of Steam Apparatus, at the 33d Congress of that society held at Paris in 1909.*

Their attention was called to this subject by the fact that several more or less discrepant reports as to the effect of age on boiler plate had been made from time to time, particularly some tests on the material of very old boilers made by Walther-Meunier, and reported in 1903–1904 to the same Association. He had found some old plate so brittle that he was of the opinion that all boilers should be retired after from 30 to 35

* Comptes Rendus Des Séances Du 33e. Congrès Des Ingénieurs en chef Des Associations De Propiétaires D'Apareils A Vapeur. Tenu a Paris, 1909.

years use, if worked 12 hours a day, and if worked 24 hours, he thought that a lower limit of useful life should be set, say 20–25 years. This raised a storm of protest and discussion among the French engineers, many of whom cited tests to the contrary, and the result was that his work came to be largely discounted because of lack of data as to the original condition of the material.

Olry and Bonnet were interested in this controversy, and when they were presented with the opportunity of testing some old boilers whose history was available, they made the investigations which form the basis of the report we are considering.

La Société des Hauts Fourneaux, Forges et Aciéries de Denain et D'Anzin, a French steel works of considerable note, installed during 1873 and 1874, 14 boilers for use at their works. They were made by Schneider et Cie, at Creusot, and were of the type illustrated in Fig. 5, cylindrical, with internal furnaces and direct tubes, surmounted by a dome. The settings were such that the products of combustion passed first through the tubes, then returned under the right-hand side of the shell to the front, where they passed across, and back to the flue, under the left-hand side of the shell. A longitudinal baffle wall for this purpose was provided under the center line of the boiler, as is indicated in Fig. 5. The boilers were designed for a pressure of 71 pounds, but were later tested and worked at 78 pounds (5.5 kg. per sq. cm.). They had a heating surface of 1,270 square feet, were oil fired, and forced day and night, except Sundays, for more than 30 years. In 1900 the rate of firing, which is typical of the service they rendered throughout their life, was such as to consume about 150 kg. of oil per square meter of grate per hour, which is equivalent to 31 pounds of oil per square foot of grate per hour, a very high rate indeed. The evaporation obtained was about 6 pounds of water per pound of fuel. The material of which the boilers were constructed was Creusot wrought-iron, designated by the following numbers: body of the boiler, No. 2; heads, lower furnace sheets, and domes, No. 4; upper furnace sheets and front tube sheets, No. 6. The entire battery was overhauled between 1905 and 1907, as the result of a general breaking down from old age, and has since been entirely replaced. This overhauling, however, gave the opportunity for obtaining test specimens, and the data given was obtained at this time.

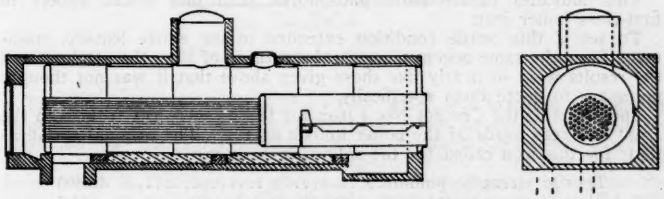

FIG. 5.—BOILER KNOWN AS S–3.

The iron known as Creusot No. 2, was made to meet the following guarantee:

| | |
|---|---|
| Tensile strength, (minimum), pounds per square inch.. | 47,330 |
| Elongation in 4 inches, per cent..... | 6.5 |
| Reduction in area, per cent..... | 6 |

The original thickness of sheet was .55 of an inch.

In January, 1905, a crack developed in one of the boilers, known as S–3, in the bottom of the third course, in the fourth girth seam, neces-

sitating the removal of a portion of the sheet. Eight specimens for tensile test of standard (eight-inch) size, were cut from this sheet and broken, with the following average results:

| | |
|---|---|
| Tensile strength, specimens cut lengthwise, pounds | 41,700 |
| Elongation (4 inches), specimens cut lengthwise, per cent. | 3.1 |
| Tensile strength, specimens cut girthwise, pounds | 39,000 |
| Elongation (4 inches), specimens cut girthwise, per cent. | 1.7 |

Fourteen specimens of the type and dimensions shown in Fig. 6 were also cut and tested for brittleness by the impact test, in which a ram or hammer, of known weight, is allowed to fall from a known height, striking the specimen fairly on its flat side, at the point of least breadth. As a sort of standard of comparison, by which one can gage the performance of the various specimens under this test, it may be well to state that a similar specimen of good modern boiler steel, ½ inch thick, is required by French standards to withstand a blow from a ram of 28.7 pounds, falling 13.12 feet (13 kilograms, falling 4 meters). This means an expenditure of 374 foot-pounds of work without starting a fracture. These particular specimens were fractured on the average, by a blow from a 26.4-pound ram falling 19.7 inches, or with an expenditure of 43.8 foot-pounds.

The deterioration of the material as indicated by these tests was so great that another group of specimens was cut from the same shell, yielding the following average results:

| | |
|---|---|
| Tensile strength (long.), pounds | 38,400 |
| Elongation, (8 inches), (long.), per cent | 2 |
| Elongation, (8 inches), (trans.), | Practically nothing. |
| Impact | Complete fracture, 28.7 pounds, falling 19.7 inches. |

A chemical analysis showed the following composition:

| | |
|---|---|
| Carbon, per cent | 0.07 |
| Manganese, per cent | 0.05 |
| Sulphur, per cent | 0.046 |
| Phosphorus, per cent | 0.290 |

This indicates rather more phosphorus than one would expect in first-class boiler iron.

To see if this brittle condition extended to the entire battery, specimens from the same region were cut from three of the other boilers, and the results were so nearly like those given above that it was not thought necessary to quote them specifically.

Specimens of the Creusot No. 4 iron for testing were cut from both the front and rear heads of the boiler known as S–4. The original specifications for this iron called for the following properties:

| | |
|---|---|
| Tensile strength, pounds | 48,800 |
| Elongation, per cent | 14.6 |
| Reduction in area, per cent | 1.3 |

The metal as tested from the front head of S–4, gave values for these quantities as indicated below:

| | |
|---|---|
| Tensile strength, pounds | 43,400<br>40,700<br>41,300 |
| Average, pounds | 41,800 |
| Elongation (4 inches), per cent | 10<br>11<br>18 |
| Average, per cent | 13 |
| Reduction in area, average, per cent | 2.2 |

A weight of 37.45 pounds falling 39.37 inches started a fracture, while modern steel of this thickness would be required to withstand the impact of a like weight falling 13.12 feet, without injury.

The specimens of the same (No. 4) iron from the rear head of boiler S-4 gave the following results:

| | |
|---|---|
| Tensile strength, pounds | 44,500<br>43,400<br>45,700<br>46,000 |
| Average, pounds | 44,900 |
| Elongation (4 inches), per cent | 17<br>13<br>11<br>10 |
| Average, per cent | 12.75 |
| Reduction in area, average, per cent | 1.42 |

Subjected to the impact test, 3 out of 8 specimens, failed under a blow from a ram of 44 pounds, falling 6.6 feet. Chemical analysis of the material showed its composition to be as follows:

| | |
|---|---|
| Carbon, per cent | 0.05 |
| Silicon, per cent | 0.15 |
| Manganese, less than, per cent | 0.10 |
| Sulphur, per cent | 0.010 |
| Phosphorus, per cent | 0.100 |

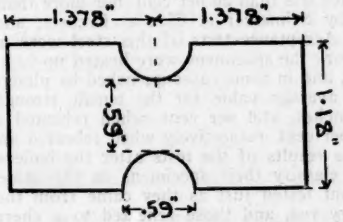


FIG. 6.—IMPACT-TEST SPECIMEN.

These tests indicate that the No. 4 iron, which was lower in phosphorus than the No. 2, had not deteriorated to so great an extent, although its condition was far from satisfactory. It is rather interesting, however, in view of the tests of steel boilers reported later, to see that the front head, which in this case was always at a moderate temperature, since the boiler was internally fired, should have deteriorated more than the rear head, exposed as it was to contact with the hot gases from the tubes.

The authors state that they were unable to report the condition of the No. 6 iron, since, due to the many repairs which had been made to the furnaces from time to time, they were unable to positively locate any of the original iron of this grade.

In the consideration of these results, giving as they did such obvious evidence of impairment, the authors raised the question as to whether the iron might be made to regain some or all of its original ductility by reworking. To this end billets were made by piling up and welding small blooms from the scraps of test pieces of each sort of iron, the billets were rolled into bars, from which new test pieces of the reworked material were cut and tested. The results are tabulated below.

Reworked No. 2 iron:
- Tensile strength, pounds..... 53,500
- Elongation, per cent..... 23
- Reduction in area, per cent..... 2.8
- Impact test.....28.66-pound ram, falling 4.88 feet, started fracture.
- Bent cold through 135 degrees.

Reworked No. 4 iron:
- Tensile strength, pounds..... 51,500
- Elongation, per cent..... 25
- Reduction in area, per cent..... 2.5
- Impact test, specimens cracked under blows from a 44-pound ram, falling 12.3 to 13.12 feet.
- Cold bend test, bent through 180 degrees without cracks of any sort.

These tables show that by reworking a most astonishing improvement in ductility was produced. The No. 4 iron became a most excellent material, equal to good boiler iron, though somewhat inferior to the best boiler steel, while even the No. 2 iron showed properties sufficiently good for many purposes, though still rather brittle for boiler use.

The steel boiler from which specimens were tested was one of a battery of 22 fire-tube boilers, with longitudinal bottom drums made by Carron-Delmotte, at Anzin, for the sugar refinery of C. Say, in Paris. The specifications called for Siemens-Martin basic steel with tensile strength greater than 51,000 pounds, and not over 56,000 pounds; elongation in 8 inches, not less than 26 per cent. nor more than 40 per cent. The steel was made by Schneider et Cie., at Creusot, and branded "A. S. acier soudable." Acceptance tests of this steel were made by Cornut in 1887. For this work the specimens were heated up to a cherry red before they were broken, and in some cases quenched by plunging them in water. He found as an average value for the tensile strength, 53,000 pounds; elongation in 8 inches, 31.6 per cent. when reheated simply, and 68,000 pounds and 18.7 per cent. respectively when reheated and quenched.

In reporting the results of the tests after the boilers had been in service the authors classify their specimens in the same manner, that is, those untreated but tested just as they came from the boiler, those reheated to a cherry red, and those reheated to a cherry red and subsequently quenched by plunging them into water maintained at a temperature of 82 degrees F.

In 1908 the first two boilers of this battery were to be removed, and the owners gave the opportunity of testing the quality of the material, as they were anxious to see if the steel had deteriorated to such an extent as to render this removal unadvisable. They accordingly gave the lower or fire sheet of the right-hand bottom drum of boiler No. 2 for the purpose. This sheet was cut up and tested through the courtesy of the steel works at Denain, who placed their equipment at the disposal of the authors. Fig. 7 will indicate the manner in which the sheet was divided, and will also serve to show how the specimens were placed with regard to the position of the sheet in the boiler.

These boilers had been in service 24 hours a day during the interval 1888–1908, with the exception of Sundays, and certain intervals for cleaning and inspection. No repairs of any moment were ever made. The records of the owner show that this particular boiler had been in service a total of 134,172 hours, consuming 3,898.13 metric tons of soft coal, and 13,050.7 metric tons of coke. This gives for the average rate of combustion, 42 kilograms per square meter of grate per hour, or in the more familiar English units, 8.6 pounds per square foot of grate per hour, certainly very moderate service. The averages of the tensile tests, classed

in groups as to their location with respect to the fire, and also divided into the three sets mentioned above, depending on the treatment they received after cutting from the sheet, will be found in the following table:

TESTS OF STEEL FROM THE FIRE SHEET OF No. 2 BOILER.

| | Untreated specimens. | Reheated specimens. | Reheated and quenched specimens. |
|---|---|---|---|
| Specimens from portion of sheet protected by front wall: | | | |
| Tensile strength ..... | 53,500 | 53,900 | none tested. |
| Elongation, per cent..... | 25.2 | 26.5 | none tested. |
| Specimens from above the grates: | | | |
| Tensile strength ..... | 53,500 | 53,800 | 70,000 |
| Elongation, per cent..... | 23.0 | 27.1 | 20.6 |
| Specimens from over bridge-wall: | | | |
| Tensile strength ..... | 51,300 | 52,900 | 70,700 |
| Elongation, per cent..... | 24.7 | 26.5 | 19.7 |
| Specimens from behind bridge-wall: | | | |
| Tensile strength ..... | 51,400 | 51,500 | 68,700 |
| Elongation, per cent..... | 24.8 | 30.2 | 21.2 |

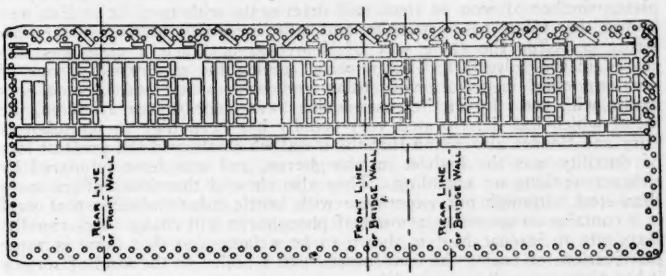


FIG. 7.—FIRE SHEET OF C. SAY ET CIE. BOILER NO. 2. SHOWING THE LOCATION OF THE SPECIMENS WITH RESPECT TO THE FIRE.

Cold bend test of 38 specimens, fairly satisfactory. The real extent of the change in this boiler steel was not disclosed until the impact tests were made. One hundred and sixty-two impact specimens were tested all told, but as 10 of these were in the nature of special tests, to determine the effect of various sorts of abuse on this steel, such, for instance, as hammering it violently when at a blue heat, they were not included in the averages, or percentages to follow. Out of the 152 tests considered to represent the real condition of the material there were 122 in which the specimens were untreated. Out of this number 58 specimens failed to pass the test expected of new steel of this class and thickness (7/16 inch) that is to stand without cracking, the impact of a 22-pound ram, falling 13.12 feet. It was found, however, that none of the reheated specimens, whether quenched or not, failed, or that, in other words, although the steel was found to have grown brittle, it could be made to fulfill the conditions of the impact test by heat treatment. If we now calculate the percentage failure, we find that based on the whole number broken, 38.1 per cent. failed, but if we consider only the untreated specimens, the percentage is seen to be 47.5 per cent. It is also of interest to

see where these specimens were located with respect to the grate, the bridge wall, etc., and to see if there is any connection between the percentage of failures, which must be taken to represent the average brittleness of the material, and the sort of treatment it received as regards temperature and heat transfer. It will be seen from the table given that there is such a connection, that it is identical with the changes in the elongation with exposure to the fire, as shown in the table of tensile tests, and that, as we should expect, the metal over the grate suffered most, that over the bridge wall next, the metal located behind the bridge less, and that in the front wall, and therefore entirely protected from the direct action of the flames, the least of all.

UNTREATED IMPACT SPECIMENS GROUPED AS TO THEIR LOCATION IN THE FIRE SHEET.

| | In front wall. | Over grate. | Over bridge. | Behind bridge. |
|---|---|---|---|---|
| Total number tested..... | 17 | 45 | 14 | 46 |
| Number of failures..... | 6 | 26 | 7 | 19 |
| Number intact ..... | 11 | 19 | 7 | 27 |
| Failures, per cent..... | 35.5 | 57.8 | 50 | 41.5 |

The work of Oiry and Bonnet shows pretty conclusively that boiler plate, whether of iron or steel, will deteriorate with use. It is also well known that tubes, tube-cap bolts, and other materials used in boiler construction suffer the same sort of depreciation. Such a case was discussed in the July, 1912, "Locomotive," giving the experience of one of our own chief inspectors with tubes and bolts which had become very brittle with use. In all the cases which have come to our attention, the metal which has deteriorated very rapidly has been high in phosphorus. Olry and Bonnet also found that the metal which showed the greatest loss of ductility was the highest in phosphorus, and was least improved by either reworking or annealing. They also showed that iron suffers more than steel, although our experience with brittle tubes indicates that steel if it contains an excessive amount of phosphorus will change very rapidly. Exposure to intense heat is shown to be a factor, so that there is some justification for basing the condemnation of a boiler on the kind of service it has given, as well as on its life.

Passing now to the hydrostatic tests first considered, in which five very similar boilers all of which had seen some thirty years of service, were tested after they had been condemned for old age by the "Hartford," it was found that all of them showed structural weaknesses, especially about the cast-iron manhole frames, which gave abundant evidence of the wisdom of their retirement from service. It is interesting in this connection to recall that all three boilers which were permitted to fail at that point (no patches being used) did so at pressures surprisingly close together, namely, 265 pounds, 260 pounds, and 270 pounds. It has been said with some emphasis, however, in the engineering press, that none of these boilers had suffered any deterioration from age.

It is true that the boilers of the Oliver Iron Mining Co. proved to be made of a material whose properties were still excellent at the time of the test. Tests of the material of the other two boilers are not yet available for discussion. The facts of the case, however, which seem to need emphasis as showing the real reasons underlying such a retirement as these boilers present, are these. The art of boiler making and designing has progressed materially in say thirty years, and the boilers of that period, if of good material, do not compare especially well as to safety with the product of the present of equal grade. It is also a matter of record that boiler steel undergoes a slow but certain loss in strength and ductility. To be sure, these changes are slower for good steel than for iron, but the

presence of even a moderate excess of phosphorus hastens the process materially.

Add to this the other equally obvious fact, that such deterioration can be detected by none of the ordinary inspection methods, and that even tensile tests may fail to indicate the extent of the change completely, and it would seem that the justice of the position which makes for old age retirements, was established beyond controversy.—"The Locomotive."

## MAKING ENGINEERING REPORTS.

By PROFESSOR H. C. SADLER, of the University of Michigan.*

The first thing I will take up is—What is the function of a report?

When an engineer is called upon to make a report about anything, the main object is to convey to the minds of the people engaging him a certain number of facts so that they may know practically as much about an investigation as if they had performed the work themselves. While it is often true that the principal is not versed in the minutiae of the problem, he generally has enough knowledge to follow a technical report with intelligence. To do this, however, he must have all the facts—that is the first point I want to make—and these must be presented in as clear and concise a form as possible. A bare statement of results of any investigation, while possessing a certain amount of value, is often practically useless unless accompanied with a more or less detailed account of the manner in which they were obtained, the conditions obtaining at the time, and the various precautions taken for insuring the greatest possible accuracy. Unless such a statement accompanies a report it is quite possible that wrong impressions may be conveyed or erroneous conclusions drawn.

In cases where reports are to be submitted to those possessing a fairly thorough knowledge of engineering in general, a certain amount of explanatory matter may be omitted, particularly if such matter is common knowledge. Nevertheless, even under the above conditions, it is usually safer rather to err on the side of elaboration than meagreness.

Aside from containing an account of facts, reports should, in the great majority of cases, also show the conclusions that may be drawn from the facts submitted.

The person immediately in charge of an experiment or investigation is usually more competent to discuss the results of his own investigations than anyone else; and it may be taken as a maxim that a discussion of results is a "*sine qua non*" to all reports.

It sometimes happens that a report upon certain facts is demanded, and an express statement made that nothing more is required. In such cases my advice is that the offer should be politely declined; for even though the intentions may be perfectly honorable, it is quite possible that erroneous conclusions may be drawn, or the report used for purposes other than those originally intended. Sometimes incalculable harm may be done, both to the person preparing the report and to other innocent persons. This brings us to the final function of a report, viz: A summary of the conclusions and recommendations, which in the mind of the person making the report are justifiable.

### IMPORTANCE OF REPORTS.

So far we have considered what may be described as the function of the technical part of a report, *i. e.*, the statement of facts and conclusions to be drawn therefrom. In a large number of cases, however, a complete

* Extracts from a lecture before the School of Marine Engineering, U. S. Naval Academy, Annapolis, Maryland.

report must go somewhat further, and contain certain definite recommendations. A good deal of care should be exercised in this part, as matters of great moment and, in many cases, large monetary considerations are involved. The conclusions should therefore be examined carefully and from all possible standpoints before the author commits himself to any definite recommendations. Even a simple investigation and test such, for instance, as that of some new alloy or metal, may have far reaching effects upon the design and construction not only of ships and their machinery, but also upon the engineering world generally.

While on this subject a slight digression may be excused, especially as it tends to emphasize the point under discussion. Few people realize the important position occupied by the engineer in the present state of social and industrial development. Without exception, all the "biggest" things in the country, such as large irrigation projects, development of water powers, railroads and all means of communication, economic problems in manufacture or, in one word, "conservation," demand the services of the engineer. In most cases, upon his word and his alone, millions of dollars are expended without question. When, therefore, we see the trust that may be placed in an engineer's report, we begin to realize the importance of the necessity for careful attentions to details and the conscientious study of results of investigations before making recommendations.

One other point deserves attention, the lack of which is only too obvious in some cases, and that is the actual composition and English of the report. While the occasion scarcely demands that a large amount of time should be spent in molding the various sentences into elegant English, nevertheless a certain amount of care should be exercised in the composition, so that the report "reads" well and conveys the intended meaning without any doubt.

Long, verbose and padded sentences should be avoided, and conciseness aimed at whenever possible, provided, of course, that this does not detract from a clear understanding of the point considered.

The principal facts should stand out clearly but without adopting too much the use of what is sometimes called "telegraphic" English.

We may now pass on to what may be called the skeleton of all reports, *i. e.*, the principal parts or main divisions common to all documents of this character. In general they may be divided into the following:

(1) Letter of transmittal.
(2) Statement or description of problem.
(3) Work accomplished.
(4) Conclusions.
(5) Diagrams, plates and illustrative matter generally.

It may be necessary to elaborate the above in certain cases, which will be discussed later, but on the whole they may be taken as representative sections.

(1) Sometimes the part that is designated "letter of transmittal" is summarized in the first paragraph of the report, but in any case the first page should contain a brief statement of the authority under which the work has been conducted, together with dates and references to any letters and instructions bearing upon the same.

(2) Statement or description of problem:

This should include a full description of the objects to be attained, the machinery used, methods adopted to obtain the desired information, etc.

(3) Work accomplished:

Under this general heading should be included a full account of the data obtained from the results of any observations, tests, calculations, etc.

(4) In drawing any conclusions from the above, it is as well to have the detailed part, with all the necessary discussion and reasons, come first;

and this should be carried out for the whole subject. At the end of this a short and concise summary is often of great value, particularly if it is properly arranged.

The plates and diagrams are generally placed at the end of the report—not interspersed with the reading matter. They should be numbered and arranged so that they follow the same sequence as the text, and, if possible, be of uniform size throughout.

### TYPES OF REPORTS.

The name of the different kinds of things upon which an engineer may be called to report is legion; but for our purposes the following may be taken as illustrations:

(*a*) A test of any kind.
(*b*) A machine or design.
(*c*) A visit to works.
(*d*) An engineering project.
(*e*) A valuation.

Of these the first three will cover the majority of cases with which we are immediately concerned. The fourth and fifth are of such large scope and cover a variety of cases that it is impossible to consider them here.

The following, however, may be taken as typical outlines to be followed in the cases considered.

### *Report on Simple (Physical) Test.*

1. Letter of transmittal.
2. Object of test and short discussion of results arrived at.
3. Description of piece or pieces tested—dimensions, physical and other properties—reference number to each piece.
4. Arrangements for test—apparatus and instruments, including method of conducting.
5. Data obtained.
6. Discussion of results (behavior of pieces) and conclusions.
7. Plates.

Sometimes 6 follows 3.

### *Report upon Machine.*

1. Letter of transmittal.
2. General description of machine, builders, when and where built. Objects for which designed. Principal dimensions, capacity, weight, etc.
3. Detailed description of parts and method of operation. Materials, general design, etc.
4. Discussion of design, analysis of operation of whole and parts.
5. Recommendations.
6. Plans, photos, etc.

Same general form for a design.

*For recommendations.*—Careful investigation into design of parts for strength, *i. e.*, sizes, should be made.

### *Report on Test of Machine.*

1. Letter of transmittal.
2. Object of test and short discussion of points to be obtained.
3. Full description of machine tested. (Plates and blue prints at back.)

4. Arrangements for test. Apparatus used. (Plate.) Calibration of instruments.
5. Method of conducting test. Different runs and conditions.
6. Data obtained. Tables. Curves, etc.
7. Behavior of machine—during test, examination after.
8. Discussion of results, and conclusions.
9. Recommendations, if any.
10. Plates, figures, etc.

Sometimes 8 and 9 are placed directly after 3, for the reason that the results and recommendations are the things in which the person reading the report is most interested. In such cases the actual data and other details are added in the form of an appendix.

*Report on Visit to Works.*

1. Letter of transmittal.
2. Name and location of plant (and main office). (Sometimes location of branch factories.)
3. Description in general of products of manufacture—Classify.
4. General layout of plant—buildings, arrangements for receiving raw material and shipping product, and handling generally.
5. Detailed description of methods of manufacture. Processes used, how applied, tools and machinery in each building (for each product).
6. Methods of inspection and tests (if any), raw material, finished product.
7. Comments.

If report on commercial and operative side, in addition:

(1) General organization. (*a*) Officers. (*b*) Commercial. (*c*) Sales. (*d*) Engineering. (*e*) Works.
(2) Method of cost keeping and tracing up of work.
(3) Financial. Capital stock; dividends.

*Valuation.*

These may vary from valuation of a simple machine to railroad with right of way, equipment, land and buildings, track, telephone and telegraph, etc. But in general the reports should cover:

1. Name and general description of property or machine, etc., where located, in case of machine, by whom built and where and when, photos, plans, etc., typical.
2. Systematic analysis of parts, with cost of reproduction, per cent. of value of this at present time, present value; use tabular form.
3. Summary of various details and values—into general groups.

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## THE CORROSION OF STEAM BOILERS.*

By L. Archbutt, F. I. C.

All natural waters when heated in wrought-iron pipes may be expected to exert a corrosive action on the iron at first, and whether the action continues will depend entirely upon whether the pipes do or do not become protected by a deposit of carbonate of lime. Galvanizing will protect the iron so long as the zinc coating lasts, which will not be for

* Abstract of paper on "The Corrosion of Iron and the Protection of Structural Iron Work," read before the Derby Society of Engineers.

long, but in the case of hard chalk waters it will probably last quite long enough to preserve the iron from corrosion until the protective coating of carbonate of lime has formed, and will, therefore, prove advantageous for such waters. Steel reports a case where Green's economizer pipes of cast iron were dissolved away by the soft water of Sydney, N. S. W., in a few months only a thin outer shell remaining. About 84.5 per cent. of the iron and the whole of the combined carbon and sulphur had disappeared, and a residue remained in the pipes containing the whole of the graphite, phosphorus and silicon, the latter as oxides, which formed a thick layer of plumbago-like material on the inside of the pipes.

It makes a great difference whether there is a continuous flow of fresh hot water through the pipes, as in hot-water service pipes, or whether there is merely a circulation of the same water, as in heating pipes. In the latter case the carbonic acid and oxygen are not renewed, and the corrosive action, which is slight, soon comes to an end. One, therefore, never hears of the corrosion of heating pipes, the life of which is indefinitely long.

In steam boilers a quite different set of conditions prevails. Here the carbon dioxide and the oxygen are rapidly expelled from the water, and their corrosive action is greatly lessened, though not entirely removed, for the feed water added continually renews these gases, and consequently they are never entirely absent, and hence their injurious effect on the iron, though more slowly, may be none the less surely produced. In boilers, however, other phases of the question present themselves, such as the corrosive action of saline solutions at high temperatures, the influence of impurities in the boiler plates, the effects of strain, and of the contact of dissimilar metals.

It is a well-known fact that all waters are not equally corrosive, and that the corrosivity depends upon the nature and amount of the dissolved salts. Very soft waters, such as that of Loch Katrine, for instance, contain such a trifling quantity of matter in solution that they deposit next to nothing on the boiler plates, which remain freely exposed to attack by the dissolved gases. The amount of corrosion that goes on is then dependent upon the amount of carbonic acid and oxygen brought in with the feed water, and also upon the composition of the steel or iron boiler plates. Most waters, however, contain salts in solution. Some of these salts, which make the water hard, *i. e.*, destructive to soap, deposit carbonate of lime, magnesia and sulphate of lime as a crust upon the plates, which protects them from corrosion less or more according to whether the incrustation is soft and porous or hard and non-porous; other salts, generally the sulphate, chloride and carbonate of sodium, do not deposit, but remain dissolved, and, as the water is evaporated, form a corrosive solution of increasing strength. Softened waters, which have had their incrusting salts removed by chemical treatment, leaving the soluble salts in solution, and perhaps even increasing their amount, are more liable to attack the plates than the hard water, and care has to be taken not to remove the whole of the incrusting salts, lest by so doing the plates should be exposed to corrosive attack, and the evils caused by excessive incrustation replaced by another and perhaps greater evil. Cribb and Arnaud in 1905, and Heyn and Bauer more recently, have shown that dilute solutions of sodium or potassium carbonate stimulate corrosion, and waters which are softened by sodium carbonate may thus be rendered actively corrosive if the sodium carbonate be used in excess. As the amount of the alkali increases, the corrosivity increases up to a certain concentration, and then diminishes again, *and we have the remarkable fact that strong solutions are protective to iron, whilst dilute solutions are actively corrosive.*

It is only within the last four years that the action of saline solutions upon iron has been systematically studied, by Heyn and Bauer in Germany

and by Friend and Brown in this country. Heyn and Bauer have studied the action at ordinary temperatures of many of the most commonly-occurring salts, in solutions ranging in strength from highly dilute to saturated solutions. Their results are capable of being represented diagrammatically by a curve. The general effect of dissolving any single salt in distilled water at ordinary temperature is to make it more corrosive to iron. If the corrosive effect of distilled water be so much the addition of a soluble electrolyte increases the action up to a further point which is called the "critical concentration," beyond this the corrosive effect diminishes as the concentration increases until it becomes *nil*, and from this up to the saturation point the solution is actually protective. In the great majority of cases, however, the *nil* point is never reached; in other words, the water becomes saturated with the salt before the corrosivity of the solution ceases, and the curve, therefore, terminates at some earlier point.

Solutions of chromic acid and chromates are exceptional, in that they have no "critical concentration," and tend to be protective from the first. Cushman has applied this fact in practice to the preservation of fence wire in America by passing it through a solution of potassium bichromate of suitable strength. He found that the wire so treated resisted rusting longer than ordinary wire, and he has even proposed to put potassium bichromate into steam boilers to prevent or check corrosion. Soon after reading his paper I made some experiments in the laboratory, and found that the addition of bichromate to a concentrated water drawn from a boiler in proportions ranging from 10 to 100 grains per gallon, did not prevent the corrosion of steel, especially when in metallic contact with copper, as in a locomotive boiler; in fact, in some of the experiments serious local pitting resulted, and I came to the conclusion that it would be dangerous to put bichromate at any rate into locomotive boilers. This subject has been more recently and completely investigated by Friend and Brown, who have shown that potassium bichromate, instead of being protective, may become actually corrosive in the presence of other salts, owing to the liberation of acids which tend to destroy the passivity produced by chromic acid. In the presence of other electrolytes much better results are obtained with neutral potassium chromate than with bichromate, but the quantity required to produce a tangible result is considerable.

One is led to enquire why the corrosive effect of dilute solutions should be lessened as the strength increases. The answer appears to be that the lessened corrosion is due to the diminished solubility of oxygen, confirming what has been known about salt solution for 70 years. Adie, in 1845, read a paper before the Institute of Civil Engineers, in which he showed that wrought iron in 80 days lost 19 times as much by corrosion in fresh water as in saturated common salt solution, and cast iron four times as much. Friend has recently shown, in fact, that in the entire absence of air, salt solutions have no action whatever upon iron. Sea water, nevertheless, is corrosive at ordinary temperature even in the absence of air, owing to the magnesium chloride present. In marine boilers, however, a certain amount of sea water is found beneficial. Rowan, in 1876, quoted the case of a steamer whose boilers were worked with a minimum proportion of sea water, and in twelve months were so badly corroded that a new set were required. In the new boilers the proportion of sea water was considerably increased, and the life of the boilers greatly prolonged in consequence. It is evidently, therefore, of the greatest importance to exclude air from boilers as much as possible, and to take care, especially in using condensed water, that air is not pumped in with the feed. The air dissolved in natural soft waters must of necessity be introduced with the feed. Attempts have been made to extract this air by passing the water through a vacuum chamber, but with no very marked success. Beneficial results have, however, resulted from the presence in the feed of organic matters which combine with the dissolved oxygen, such as tannin. No

doubt the beneficial effects of sodium arsenite in preventing corrosion are due to the same cause.

Although the systematic researches of Prof. Heyn and Dr. Friend, to which I have referred, have thrown very valuable light upon the corrosive effects of salts in solution, much more work needs to be done, especially at higher temperatures, and under conditions such as obtain in steam boilers. Most of the researches have been made at temperatures no higher than 200 degrees F., and in solutions freely exposed to air and light. If corrosion took place in steam boilers to the extent shown possible by some of these experiments, the lives of boilers would be measured by weeks or months instead of years. We know that temperature influences the results vastly, especially by reducing the solubility of gases, and also that corrosion is much less active in the dark than in daylight.

Another cause of active corrosion to which I must not omit reference is the presence in the water of fatty acids. Corrosion from this cause has become less frequent since the introduction of mineral oils as lubricants, but in the days when animal fats, such as tallow, were commonly used in cylinders, and even put directly into boilers, such cases were common. Animal and vegetable oils are compound bodies which, when heated with steam, split up into fatty acids and glycerine. The fatty acids thus set free are very corrosive to iron and many other metals. Oleic acid from olive oil, if heated in a test tube with water and iron filings, attacks the iron vigorously, forming brown ferrous oleate and evolving hydrogen gas. The ferrous oleate is decomposed, by oxidation, into ferric hydroxide and free oleic acid, which again attacks more iron, and in this way quite a small quantity of free fatty acid has been known to perforate wrought-iron plate more than ½ inch in thickness. Pitting and corrosion along the bottom of steam boilers may result from this cause, and also at the flanges and screwed joints of water pipes when red lead and linseed oil are used to make the joints. The danger of allowing grease of any kind to obtain entrance to steam boilers is now too well known to need insisting upon. In screwing the stays, fittings and wash-out plugs into boilers, pure mineral lubricants should be used exclusively, and all surplus oil should be removed from new boilers by thorough digestion under pressure with hot soda solution and washing out with clean hot water before putting the boilers into use.

It is a well-known fact that neither iron nor steel are homogeneous metals. If the polished and etched surface of a piece of wrought iron be highly magnified it is seen to be composed of polygonal crystals of iron and particles of slag, and if you look at an etched specimen of wrought iron you will see that it is built up of layers of iron, between which the layers of slag are sandwiched. When medium carbon steel is similarly magnified it is seen to have a still more complex structure, composed of areas of iron containing a solution more or less manganese, phosphorus and silicon, darker areas of carbide of iron occurring in the complex which we call pearlite, sulphur existing as isolated particles of sulphide of manganese and frequently particles of slag. These different constituents are not uniformly mingled together, but present a variety of structures depending upon the heat treatment and kind of work the steel has undergone. The pearlite, for instance, is not always laminated. In unhardened steels it is more or less completely emulsified, and in hardened steels granular. According to a recent report of the B. A. Committee, which is investigating the corrosion of carbon steels, the state of division of the carbon has a great influence on the rate of corrosion in sea water, the hardened steels corroding more rapidly than the unhardened or tempered steels. The different varieties of pearlite also differ in polarity. Moreover, the impurities in the molten steel have a tendency to segregate during solidification, sometimes so seriously that drillings taken from different parts of the finished material may have a quite different chemical

composition. Areas of different electrical polarity are thus formed, between which action readily occurs, stimulating corrosion. Owing to the ingenuity of Drs. Cushman and Walker we have been put in possession of a chemical reagent, to which they have given the name of "ferroxyl," which shows this in a very beautiful manner. It is a jelly containing in solution phenolphthalein and potassium ferricyanide. The phenolphthalein gives a pink color in the presence of hydroxyl ions and the ferricyanide gives a deep blue precipitate in the presence of soluble ferrous iron. The jelly merely serves to localize these effects and prevent them from running into one another. If a piece of iron or steel is placed in the melted jelly and the jelly allowed to set, there are developed on the surface of the metal pink areas and blue areas, indicating the positions of nodes of opposite polarity. The blue color shows the anodic areas where iron is going into solution, the pink color shows the cathodic areas where no iron is being dissolved. It is thus possible to demonstrate in a simple and convincing manner a fact which throws a flood of light on one cause of localized corrosion or pitting, especially in boilers.

We learn from the ferroxyl indicator the great importance of obtaining for such purposes as steam boiler construction iron and steel as homogeneous as possible, and it would be worth while to pay a high price for such material. As Cobb has stated in a recent paper, the presence of an impurity in iron determines so many corrosion centers, and its influence depends more on quality and distribution than on quantity; a more homogeneous iron, even if chemically less pure than another, may, therefore, be more highly resistant to corrosion. The same difference of electrical polarity as is found between different areas of the same piece of iron exists between different pieces, every piece tested showing electrical effects with every other. Not only, therefore, is homogeneity of composition important in each individual part of a structure exposed to corrosive influences, but the different parts should be as much alike as possible.

Owing to the extreme difficulty of preventing segregation of the impurities in iron and steel and of producing a perfectly homogeneous material, the less impurities the better. Unfortunately, the nearer a commercial iron or steel approaches purity the more difficult it is to produce it in a thoroughly deoxidized and degasified condition. Cushman states, however, that there is now being made in the United States in several rolling mills an extremely pure ingot iron containing less than 0.05 per cent. of total impurities. This, he says, is no chemical curiosity, but is manufactured in basic open-hearth 50-ton converters. The metal is stated to possess a very high degree of malleability and ductility, and to be very resistant to corrosion. This pure iron is well worth the attention of engineers. One of the most injurious impurities in steel is sulphur and one of the most active in causing corrosion. In all well-made steel the sulphur exists as manganese sulphide, disseminated throughout the metal in the form of more or less minute particles which can readily be seen when a polished section is examined under the microscope. These form active corrosion centers, which are not electrolytic, but lead to the formation of sulphuric acid by oxidation, which attacks the iron. An interesting example has been given by Huntley. Serious pitting had started in a stand-by boiler at the generating station of the London Electricity Supply Corporation, which caustic soda was found powerless to stop. On examining the boilers numerous blisters were seen, varying in size up to 30 mm. diameter, the bulk of them being just below the water level. On pricking the blisters, each was found to contain a liquid with a fine black powder in suspension, and a pit was seen to be forming in the center of each blister. Analysis showed that the liquid was a solution of ferrous sulphate containing free sulphuric acid, although the water in the boiler was alkaline and contained much less sulphur than the liquid in the blisters. The pitting was traced to the sulphide of manganese in the

steel, which by oxidation gave rise to free sulphuric acid which attacked the iron, forming ferrous sulphate. This being oxidized to ferric sulphate by the oxygen in the water would attack more iron, and in the quiescent water of the stand-by boiler the oxides precipitated by the alkali in the water would not be washed away, but would seal in the corrosive liquid with a semi-permeable membrane through which the oxygen of the water could pass but not the caustic soda. The use of sodium arsenite instead of caustic soda completely stopped the trouble, the arsenite presumably using up the oxygen in the water.

The influence of strain on corrosion must be mentioned. It is a subject on which the late Thomas Andrews worked for many years, and his results are to be found in papers communicated to the Institute of Civil Engineers. He found that strain always causes a difference of potential, and the use of strained material should, therefore, be avoided where corrosion is likely to be set up by it. To this Friend attributes the unusual tendency to corrosion shown by indentations and abrasions on the surfaces of boilers.—"The Steamship."

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## THE REPORT ON THE *TITANIC* INQUIRY.

The Report of the British Court of Inquiry upon the loss of the *Titanic* was read by Lord Mersey at a final sitting held on the 30th of July. The Court found that the loss of the ship was due to collision with an iceberg, brought about by the excessive speed at which the ship was being navigated. This finding and the Report which followed were concurred in by Lord Mersey and all the five assessors. The Report has since been published as a Government paper, and forms a lengthy document of 74 pages. In the following abstract attention is confined to such portions as are of technical interest.

The Report first contains a detailed description of the ship and her machinery, which, in view of the very complete information regarding the vessel which has already appeared in "The Shipbuilder," it is unnecessary to repeat here. An account is then given of the ship's journey across the Atlantic. The Report states that the masters of vessels belonging to the White Star Line were not given any special "sailing orders" before the commencement of any particular voyage. The recognized routes were followed; and when there was any deviation from them in the interests of safety, the master's action was always approved by the company. Further, every master on his appointment was impressed with the importance of a cautious and ever-watchful system of navigation, dismissing all idea of competitive passages, and submitting to loss of time or other inconvenience rather than incur the slightest avoidable risk. The route followed by the *Titantic* was about 25 miles south of the edge of the area where field ice is seen under the usual conditions at the time of the year, but from 100 to 300 miles north of the most southerly points at which icebergs might be seen.

The Report then records the ice messages received from various vessels. Three messages, those from the *Caronia*, *Baltic* and *Californian*, were delivered to the bridge. Two important warnings which might have influenced the navigation of the vessel—those from the *Amerika* and the *Mesaba*—appear never to have been heard of by anyone on board the *Titanic* outside the Marconi room. It was urged on behalf of the owners that the vessel had been carefully and successfully navigated to avoid the ice of which she received warning. The evidence of a large number of

witnesses showed, however, that the nearness of ice was realized; and, this being so, Lord Mersey states it appears to him to be of little importance to consider whether the master had by design or otherwise succeeded in avoiding the particular ice indicated in the three messages received by him.

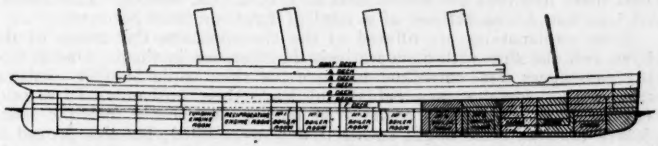
It was given in evidence that the speed of the *Titanic* up to the time of the collision was about 22 knots. The weather was perfectly clear and fine, there was no moon, the stars were out, and there was not a cloud in the sky. There was, however, a drop in temperature of 10 degrees in slightly less than two hours. This was not necessarily an indication of ice, although Sir Ernest Shackleton was of opinion that under the conditions prevalent at the time he would consider he was approaching an area which might have ice in it.

As regards the action that should have been taken, the Commissioner says: "The question is, what ought the master to have done? I am advised that, with the knowledge of the proximity of ice which the master had, two courses were open to him. The one was to stand well to the southward, instead of turning up to a westerly course; the other was to reduce speed materially as night approached. He did neither. Why, then, did the master persevere in his course and maintain his speed? The answer is to be found in the evidence. It was shown that for many years past—indeed for a quarter of a century or more—the practice of liners using this track, when in the vicinity of ice at night, had been in clear weather to keep the course, to maintain the speed, and to trust to a sharp lookout to enable them to avoid the danger. This practice, it was said. had been justified by experience, no casualties having resulted from it. I accept the evidence as to the practice and as to the immunity from casualties which is said to have accompanied it. But the event has proved the practice to be bad. Its root is probably to be found in competition and in the desire of the public for quick passages, rather than in the judgment of navigators. But, unfortunately, experience appeared to justify it. In these circumstances, I am not able to blame Captain Smith. He had not the experience which his own misfortune has afforded to those whom he has left behind, and he was doing only that which other skilled men would have done in the same position. The evidence shows he was not trying to make any record passage, or indeed any exceptionally quick passage. He was not trying to please anybody, but was exercising his own discretion in the way he thought best. He made a mistake, a very grievous mistake, but one in which, in face of the practice and past experience, negligence cannot be said to have had any part. It is, however, to be hoped that the last has been heard of the practice, and that, for the future, it will be abandoned for what we now know to be more prudent and wiser measures. What was a mistake in the case of the *Titanic* would, without doubt, be negligence in any similar case in the future."

The Report then describes the collision. Almost simultaneously with the reception of the signal and message from the crow's nest "Iceberg right ahead," Mr. Murdoch, the officer of the watch, gave the order "Hard a-starboard," and immediately telegraphed down to the engine-room "Stop; full speed astern." The helm was already "hard over," and the ship's head had fallen off about two points to port, when she collided with the iceberg well forward on her starboard side. From various experiments subsequently made with the *Olympic*, it was found that traveling at the same rate as the *Titanic* about 37 seconds would be required for the ship to change her course to this extent after the helm had been put hard a-starboard. In this time the ship would travel about 466 yards; and, allowing for the few seconds that would be necessary for the order to be given, it may be assumed that 500 yards was about the distance at which the iceberg was sighted.

Without implying that those actually on duty were not keeping a good lookout, in view of the night being moonless, there being no wind, and perhaps very little swell, and especially in view of the high speed at which the vessel was running, it is not considered that the lookout was sufficient. An extra lookout should, under the circumstances, have been placed at the stem head, and a sharp lookout should have been kept from both sides of the bridge by an officer. That it was quite possible on this night, even with a sharp lookout at the stem head, crow's nest, and on the bridge, not to see an iceberg 500 yards away, is shown by the evidence of Captain Rostron, of the *Carpathia*, who narrowly missed running his own ship into a berg. The Report states that the use of binoculars is not usual or necessary. Searchlights may at times be of service. The evidence before the Court does not allow of a more precise answer.

The collision, which took place at 11:40 P. M. on Sunday, 14th April, caused damage to the bottom of the starboard side of the vessel for a length of about 300 feet, at about 10 feet above the level of the keel, the compartments in which there was damage being indicated on the diagram. Ten minutes after the collision the water seems to have risen to about 14 feet above the keel in all these compartments except No. 5 boiler room, where it was held for some time by the forward bunker bulkhead. In No. 4 boiler room there was no indication of damage until 1 hour and 40 minutes after the collision, when small quantities of water were observed to be coming in underneath the floor in the forward part. There, was no damage whatever abaft No. 4 boiler room.



"TITANIC": DIAGRAM SHOWING DAMAGED COMPARTMENTS.

The later stages of the sinking cannot be stated with any precision. The ship's stern gradually rose out of the water until she eventually attained the perpendicular, but she did not break in two. The ship disappeared at 2:20 A. M.

The Commissioner then proceeds to some observations on the watertight sub-division of the ship. Her bulkheads were spaced to enable her to remain afloat with any two compartments in communication with the sea. Even with the four forward compartments flooded she would have remained afloat, but she could not remain afloat with the four forward compartments and the forward boiler room (No. 6) also flooded. This being the case, the question of the damage to No. 5 boiler room is only of importance in dealing with the question of what would have happened to the ship had she been more completely sub-divided. It was stated in evidence that if No. 4 boiler room had not been damaged, or had only been damaged to an extent within the power of the pumps to keep under, then, if the bulkheads had been carried to *C* deck, the ship might have been saved. The same result would have been attained if the bulkhead deck had been a watertight deck. If, however, No. 4 boiler room had also been flooded, the ship would not have remained afloat, unless, in addition to making the bulkhead deck watertight, the transverse bulkhead abaft of No. 4 boiler room had been carried up to *D* deck. To make the bulkhead deck effectively watertight for this purpose, it would have been necessary to carry watertight trunks round all the openings on the bulk-

head deck up to *C* deck. With regard to the advisability of fitting a watertight deck below the water line, there are many considerations involved, and the Commissioner recommends that this matter should be dealt with by the Bulkheads Committee for ships in general. The same remark applies to the questions of longitudinal sub-division and of extending the double bottom up the sides.

The Commissioner states that there does not seem to have been any difficulty in working the watertight doors of the *Titanic,* but that the question of their efficiency appears to have very little bearing upon the loss of the ship. Indeed, it is probable that the life of the ship would have been lengthened somewhat if these doors had been left open, for the water would have flowed through them to the after part of the ship, and the rate of flow of the water into the ship would have been for a time reduced, as the bow might have been kept up a little by the water which flowed aft. In other cases of damage to ships constructed like the *Titanic,* however, it is probable that the efficiency of the closing arrangements of the watertight doors may exert a vital influence on the safety of the ship. This, again, is a matter which should be referred to the Bulkheads Committee, with a view to their suggesting in detail where doors should or should not be allowed, and the type of door which should be adopted in the different parts of ships.

The next section of the Report deals with the saving and rescue of those who survived. The *Titanic* was provided with fourteen lifeboats, two emergency boats, and four Engelhardt or collapsible boats, or in all twenty boats, having total accommodation for 1,178 persons. It is noted that there had been no proper boat drill or a boat muster. The number of lives saved was 711 out of a total of 2,201, or 32.30 per cent.

Some explanations are offered of the circumstances that many of the boats left the ship with comparatively few persons in them. One is that the passengers were unwilling to leave the ship, while another explanation is that some women refused to leave their husbands. It is said, further, that the officers engaged in putting the people into the boats feared that the boats might buckle if they were filled; but this proved to be an unfounded apprehension, for one or more boats were completely filled and then successfully lowered into the water.

The Court was invited by the Board of Trade to report upon the Rules and Regulations made under the Merchant Shipping Acts, 1894–1906, and the administration of those Acts and of the Rules and Regulations. Charges were made against the Board of Trade during the progress of the Inquiry of a two-fold kind. First, it was said that the Board had been negligent in that they had failed to keep up to date their Rules and Regulations relating generally to the provision of life-saving appliances at sea, which was reduced in the course of the inquiry to a charge of neglect to keep the Board's scale for the provision of lifeboat accommodation up to date; and, secondly, it was said that their officials had, in the particular case of the *Titanic,* failed to exercise due care in the supervision of the vessel's plans and the inspection of the work done upon her. The evidence with regard to the first charge is examined in the Report in detail, and the Commissioner states that the outstanding circumstance in it is the blamable omission of the Board of Trade during so many years to revise the Rules of 1894 relating to the provision of boat accommodation. Lord Mersey is, however, doubtful whether, even if the Rules had been revised, the change would have been such as to have required more boat accommodation than that which was actually provided on the *Titanic;* and it is not to be forgotten that the *Titanic* boat accommodation was utilized to less than two-thirds of its capacity. These considerations nevertheless afford, in the Commissioner's opinion, no excuse for the delay of the Board of Trade.

The gross tonnage of a vessel is not considered a satisfactory basis on which to calculate the provision of boat accommodation. The Report states that much more is to be said in favor of making the number of lives carried the basis, and of providing boat or raft accommodation for all on board. Rule 12 of the Life-saving Appliances Rules of 1902, which deals with watertight compartments and boat accommodation, ought to be abolished. The provision of such compartments is of supreme importance, but it is clear that it should not be sought at the expense of a decrease in boat accommodation. When naval architects have devised practical means for rendering ships unsinkable the question of boat accommodation may have to be reconsidered; but until that time arrives boat accommodation should, where practicable, be carried for all on board.

The investigation of the second charge only served to show that the officials of the Board of Trade had discharged their duties carefully and well. The *Titanic* was efficiently designed and constructed to meet the contingencies which she was intended to meet. The bulkheads were of ample strength. They were sufficiently closely spaced and were carried up in the vessel to a height greater than sufficient to meet the requirements of the 1891 Bulkheads Committee. But the ship could have been further sub-divided so that she would probably have remained afloat longer than she did. The Board of Trade have, however, apparently no power to exercise any real supervision in the matter of sub-division. The Commissioner recommends that the Board should be empowered to require the production of all designs of all passenger steamers at an early period of their construction, and to direct such alterations as may appear practicable for the purpose of securing proper watertight sub-division according to the requirements of the new Bulkheads Committee when their report is available.

The Report concludes with a summary of the recommendations of the Court. Those concerning watertight sub-division and the number of boats have already been referred to. Other recommendations are that one or more of the boats should have some form of mechanical propulsion; the manning of the boats and boat drill should receive more attention than at present; lookout men should undergo a sight test; in all such ships a police system should be organized to secure proper control in times of emergency; a continuous service of wireless telegraphy should be secured by day and night; instructions should be given in all steamship companies' regulations that, when ice is reported in or near the track, the ship should proceed in the dark hours at a moderate speed, or alter her course so as to go well clear of the danger zone; the attention of masters of vessels should be drawn to the fact that it is a misdemeanor not to go to the relief of a vessel in distress when it is possible to do so; and an international conference should be called to consider and, as far as possible, agree upon a common line of conduct in respect to matters relating to the safety of life at sea.—"The Shipbuilder."

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## SMOKE INSPECTION.

A politician had his greenhorn cousin appointed a smoke inspector, says the "Kansas City Journal." The "inspector's" first monthly report read as follows:

"I certify that I have inspected the smoke of this city for the thirty days past. I find plenty of smoke and apparently of good quality. Respectfully submitted."—"Power."

## NAVAL ADMINISTRATION.

The series of papers just issued by the Admiralty indicate that in future the complicated work carried on at Whitehall will be more than ever specialized. The business of the Navy is to be distributed so as to gain the fullest advantage from the expert knowledge of the various members of the administration—naval and civil. The First Lord will, of course, continue to be the representative of the Cabinet and of Parliament, exercising a general control over all departments. Under his authority the Sea Lords will deal with naval matters and the civil element with the ordinary administration, while the Additional Civil Lord will act as buyer and business manager. The control over expenditure will be strengthened by the establishment of a permanent Finance Committee, under the presidency of the Parliamentary and Financial Secretary. The position of Controller of the Navy ceases to exist. In future the Third Sea Lord will be responsible for the study of naval science in its bearing upon material and for the creative and original task of design of ships of war—both of the sea and the air—and their armament and equipment. He will have under his superintendence all the departments which provide the Navy with ships, guns, torpedoes, armor and machinery, and will also be responsible for the evolution of aerial craft suited to the purposes of the Navy. The Additional Civil Lord—a position created in the spring and filled by Sir Francis Hopwood, formerly of the Board of Trade—will deal with all contracts for material for the Fleet, and will act, in fact, as the business member of the Board.

### A DIRECTOR OF NAVAL EQUIPMENT TO BE APPOINTED.

The Third Sea Lord will in future have the assistance of an officer to be known as the Director of Naval Equipment. He will in effect supervise, from the naval point of view, the equipment of ships under construction, and deal with technical questions relating to the repair and refit of completed sea-going ships. While relieving the Third Sea Lord of matters of contract and providing him with an assistant, it is intended, since a warship is primarily a gun-platform, to place under his orders the Department of Naval Ordnance and Torpedoes, hitherto subject to the authority of the First Sea Lord. In reference to this change Mr. Churchill remarks that the association of the Department of Naval Ordnance with the Constructive Departments has, of course, in practice been very close. It must now become absolute; and the genesis of the ideas which govern design must be identical and simultaneous with that which governs the character of weapons and projectiles. The Department of Naval Ordnance and Torpedoes will, therefore, be placed under the Third Sea Lord so far as matériel is concerned. But the First Sea Lord will be responsible for the systems of gunnery and torpedo exercises prevailing in the Fleet, the tactical employment of aircraft, and all military questions connected with the foregoing.

### DUTIES OF THE NEW CIVIL LORD.

The duties of the Additional Civil Lord comprise contracts of all kinds for the matériel of the Fleet, including ships and their machinery, armor, naval ordnance and gun-mountings, aeroplanes, airships, works, yard machinery and stores of all descriptions; also contract arrangements in connection with the disposal, salvage or loan of vessels or stores. Secondly, the general organization of dock-yards, including the provision of labor and plant, and all business questions in connection with the building and repair of ships and their machinery, whether in the dock-yards or in private yards. The Department of Contract and Purchase and the Department of the Director of Dockyards will be placed under the superintend-

ence of the Additional Civil Lord. General labor questions, including annual petitions, will, however, remain under the Financial Secretary. It is right that labor conditions should be periodically surveyed from a standpoint not exclusively concerned with the business administration of the dockyards; and the present holder of the office of Financial Secretary has, besides, special knowledge and aptitudes which fit him for this work.

### REASON FOR THE CHANGES.

In a memorandum Mr. Churchill explains the reason for these changes. The main end in view has been the reorganization of the work of the naval officer who hitherto has been at one and the same time Third Sea Lord and the Controller. Mr. Churchill recalls that prior to the appointment of the Additional Civil Lord the work of the Controller had hitherto comprised three principal spheres: First, the designing of the Fleet; secondly, the administrative construction, equipment and repair of the Fleet; and, thirdly, the great group of contract, business and financial questions arising from the second. All these functions are of high importance, and all are intimately related, but their character is distinct. The qualifications which fit an officer for the discharge of the duties connected with any one of these groups are quite different from those required for the others, yet the direct responsibility for any one of them is sufficiently important and extensive to occupy one man's time.

### RELIEVING THE FIRST SEA LORD.

The first essential has been to set the Third Sea Lord, the officer charged with the supervision of design, free from the complicated contract and financial questions which arise from the construction and repair of the Fleet, and from the business management of the dockyards. The duties assigned to the Additional Civil Lord will eventually relieve him in that respect. But besides these, he has been burdened by an enormous day-to-day administration connected with the construction, equipment, repair and refit of ships. These duties cannot in principle be dissociated from him. The Third Sea Lord must exercise a general and covering superintendence over the whole region of matériel. But if he is to be free to devote his mind to the progress of naval science and the designing of new ships of all kinds, he must be relieved in practice of these multifarious administrative duties.

### SHIPBUILDING PROGRESS TO BE WATCHED BY HIGH NAVAL AUTHORITY.

It may be taken for granted that the designs passed by the Board will, under the existing system of supervision, be correctly executed by the constructive departments or private firms. But the delays which have recently occurred over so large an area of naval shipbuilding show that the work must be continually watched by high naval authority, in respect not only of its design, but of its progress. In this field a large number of important questions of a practical nature are constantly arising, which, under the old system, fell upon the head of the Controller's Department to decide. Refits and repairs not affecting design again require no attention from the Third Sea Lord. But, on the other hand, the whole work of keeping the Fleet in good repair and getting the ships back to sea demands the constant supervision of a naval officer of considerable standing. Another long series of naval questions of a practical nature are continually arising from this great business, and must be settled easily and quickly as they come. No civilian can deal with such matters satisfactorily, and the Third Sea Lord is already fully occupied; for the new guiding principle is to concentrate his attention upon the creative and original task of design, and to free him from administrative distractions. —"Page's Weekly."

## THE EFFECT OF TEMPERATURE ON TENSILE TESTS OF METALS.

### *The Effect of Temperatures Higher than Atmospheric on Tensile Tests of Copper and its Alloys, and a Comparison with Wrought Iron and Steel.**

By Professor A. K. HUNTINGTON, Assoc. R.S.M., F.I.C., Vice-President.

The tests included in this paper were made at various times between 1900 and the present year. As the method of heating the bars to the required temperature is believed to be different to any method hitherto described, and as it has been proved to be easy to carry out, and to give reliable results, a detailed description of it may prove of interest and practical value.

One point which has been found of considerable importance in this method of testing is that the elastic limit or the yield-point can be taken, which is not the case when a heated bath or an electrically-heated jacket is used.

As a result of having to consider generally only cold tests of iron and steel, engineers are in the habit of reckoning the elastic limit as 50 per cent. of the breaking load. It will be seen from the curves given in this paper that this method of estimating the elastic limit is not correct for iron and steel at temperatures above the normal, nor, as a rule, for other metals at any temperature.

#### METHOD BY WHICH THE TENSILE TESTS AT TEMPERATURES HIGHER THAN THE NORMAL WERE MADE.

The tests were made with a horizontal machine of the Kirkaldy type. This machine is provided with a saddle which can be moved along the bed of the machine by hand-wheel or hydraulic pump. Attached respectively to the saddle and to the levers actuating the beam are two massive heads, *a*, Fig. 1, in each of which there is a vertical slot, the slot being enlarged towards its inner end, so as to form two vertical shoulders facing inwards, *b*. The test bar *c* is screwed at each end into a cylindrical holder *d*, the other end of which is screwed on to the end of a large bolt *e*, which passes loosely through a hole in a rectangular block *f*. These blocks fit loosely in the enlarged ends of the slots in the heads. When testing at high temperatures sheet asbestos may be inserted between the blocks and the heads.

The test bars are of two patterns:—(*a*) For use with high-temperature thermometers; (*b*) for use with a thermo-electric couple.

The pattern (*a*) is 18 inches long and 1 inch in diameter, except at the middle, which is turned down to 0.5 inches diameter over a length of 2 inches. The ends are screwed for about 2 inches to fit the cylindrical holders, and at 0.5 inch from each shoulder a hole is bored centrally (Fig. 2) 5/16 inch in diameter and 0.75 inch deep, to hold a thermometer. In order that the thermometer may be sufficiently enveloped by the bar, the drill is driven in until the point bulges the metal at the opposite side of the bar. Connection between the thermometer bulbs and the test bar is ensured by means of a small quantity of a low-melting-point alloy.

The pattern (*b*) is 5 inches to 6 inches long and 1 inch in diameter, the middle being turned down to 0.8 inch over a length of 2 inches. About 1 inch at each end is screwed to fit the cylindrical holders.

In order to attach the thermo-couple to the bar, a piece of asbestos sheeting, 1 inch long and ½ inch wide, is split down a little more than half

* Paper read before the Institute of Metals, September 25, 1912.

way, and a small hole made through one side about the middle. The junction of the couple is inserted through the hole from the inside, Fig. 3 (*a*), so that the small knob is just protruding on the outside. The asbestos and couple are placed on the bar with the junction of the couple

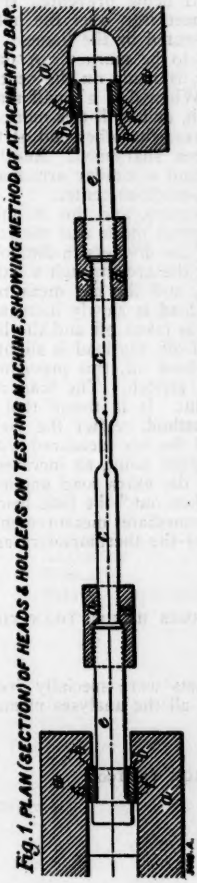


Fig. 3. SHOWING METHOD OF ATTACHMENT OF WIRES OF THERMO-COUPLE TO BAR.
(a) Couple inserted in prepared asbestos sheet.
(b) Bound with thin iron wire in position.

FIG. 4.—BAR BURNER.

touching the middle of the bar, and bound firmly into position by two or three pieces of thin wire, Fig. 3 (*b*). The curves, more particularly for electrolytic copper, copper-tin and copper-aluminum, show that very good

results can be obtained with high-temperature thermometers; but it is a saving of time to use an electrical pyrometer, which has also the important advantage of enabling bars of a larger diameter to be tested.

The bar is heated by placing a Bunsen burner under each end of the bar with pattern (*a*), and by means of a "bar burner" with pattern (*b*), Fig. 4. The burner is 14 inches long, and made by Messrs. Fletcher, Russell & Co., the part under the test bar itself being prevented from lighting by placing two overlapping pieces of sheet iron over the holes, so that only the cylindrical holders are actually heated by the flame.

Before placing the bar in the machine it has to be marked. A dark streak, about ⅜ inch wide, of some suitable ink, is made on the central portion of the bar from shoulder to shoulder. When dry, a small mark is made near one shoulder with a fine center punch, and with this mark as center and a 2-inch radius, an arc is scratched across the other end of the ink streak, using spring dividers which have been sharpened. Another punch-mark is made on the middle of the arc, and a similar arc, made at the first end of the bar, using the second punch-mark as center.

When the bar has reached the required temperature, and this is quite steady, the length between the left-hand center punch mark and the corresponding arc is measured by placing one point of the dividers in the hole, and laying the other point on the arc and viewing the arc through a hand lens. A suitable load is then placed on the bar, and the bar measured with the load on. If no stretch is observed, the load is slowly increased until a distinct stretch is observed, when the load is taken off and the bar measured. If no stretch is observed with the load off, the load is slightly increased, and the bar again measured with the load off, this procedure being repeated until the bar shows a permanent stretch. The load required to produce this is taken as the yield point. It is found that a stretch of 0.003 inch can be observed by this method. After the yield point is reached, the load is increased slightly and the bar measured with the load off. If the previous load was really the yield point, an increased stretch will be observed out of all proportion to the extra load applied. If this observation is satisfactory, the bar is "broken out," the load being increased rapidly until the bar breaks, no intermediate measurements usually being taken. With test bars of pattern (*a*) the thermometers are removed before "breaking out" the bar.

### TABULATED RESULTS OF TENSILE TESTS AT TEMPERATURES HIGHER THAN THE NORMAL.

The metals and alloys used in the following tests were specially free from impurities and were annealed. A "trace" in all the analyses means less than 0.005 per cent.

#### TABLE I.—ELECTROLYTIC COPPER, MARCH 13, 1906.

#### ROLLED 1-INCH ROD.

*Analysis.*

| | |
|---|---|
| Arsenic | trace |
| Antimony | nil |
| Bismuth, per cent | 0.0005 |
| Selenium | trace |
| Tellurium | trace |

*Tensile Tests.*

All test bars annealed 2 hours at 600 degrees C.

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 55 | 13 | 2.86 | 13.97 | 56.0 | 66.3 |
| 200 | 93 | 2.58 | 12.41 | 56.0 | 68.6 |
| 300 | 148 | 2.64 | 11.46 | 55.0 | 67.44 |
| 400 | 204 | 2.85 | 10.55 | 53.0 | 66.3 |
| 450 | 232 | 2.29 | 9.61 | 49.5 | 61.5 |
| 650 | 343 | 2.30 | 6.91 | 25.0 | 28.2 |
| 750 | 399 | 2.33 | 6.42 | 27.0 | 27.16 |
| 850 | 455 | 1.71 | 5.43 | 21.0 | 26.0 |
| 900 | 482 | 1.42 | 5.13 | 20.5 | 29.4 |
| 900 | 482 | 2.00 | 5.17 | 23.5 | 25.1 |
| 950 | 510 | 1.43 | 4.17 | 23.0 | 26.0 |
| 1,000 | 538 | 0.5 | 3.7 | 32.0 | 45.2 |

TABLE II.—ARSENICAL COPPER, DECEMBER 21, 1900.

ROLLED 1-INCH ROD.

| *Analysis.* | Per. cent. |
|---|---|
| Arsenic | 0.234 |
| Antimony | nil |
| Bismuth | 0.009 |
| Selenium and tellurium | trace |

*Tensile Tests.*

All test bars annealed 2 hours at 690 degrees C.

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 100 | 37.7 | 2.27 | 13.75 | 50.5 | 64.0 |
| 200 | 93.3 | 2.27 | 12.95 | 57.5 | 70.8 |
| 300 | 150.0 | 2.27 | 11.81 | 51.5 | 70.8 |
| 400 | 204.0 | 1.98 | 11.07 | 49.0 | 78.8 |
| 500 | 260.0 | 1.98 | 9.71 | 50.0 | 66.3 |
| 600 | 316.0 | 1.98 | 9.32 | 47.5 | 58.5 |
| 700 | 371.0 | 1.98 | 7.52 | 16.0 | 26.0 |
| 800 | 427.0 | 1.70 | 5.37 | 10.0 | 15.3 |

TABLE III.—COPPER-TIN, FEBRUARY 9, 1900.
ROLLED 1-INCH ROD.

| *Analysis.* | Per cent. |
|---|---|
| Copper | 97.673 |
| Tin | 2.408 |
| Lead | 0.024 |
| Nickel | trace |
| Iron | trace |
| Manganese | nil |

*Tensile Tests.*

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 60 | 15.5 | 4.83 | 19.13 | 56.0 | 75.0 |
| 400 | 204.0 | 2.84 | 16.77 | 53.0 | 68.6 |
| 500 | 260.0 | 2.84 | 16.25 | 45.0 | 56.4 |
| 600 | 316.0 | 2.84 | 14.54 | 23.5 | 26.0 |
| 700 | 371.0 | 2.55 | 11.36 | 21.0 | 22.5 |
| 800 | 427.0 | 3.12 | 12.97 | 36.5 | 29.4 |
| 870 | 465.0 | 2.84 | 12.31 | 38.5 | 42.2 |

TABLE IV.—COPPER-NICKEL, JULY 23, 1909. ROLLED 1-INCH ROD.

Nickel about 12 per cent.

*Tensile Tests.*

All test-bars annealed ½ hour at 650 degrees C.

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 55 | 13.0 | 9.3 | 18.1 | 55.5 | 63.0 |
| 200 | 93.3 | 9.25 | 16.2 | 55.5 | 79.0 |
| 300 | 150.0 | 9.0 | 15.1 | 54.5 | 79.5 |
| 350 | 176.7 | 7.6 | 14.1 | 57.0 | 77.0 |
| 400 | 204.0 | 7.4 | 14.1 | 55.0 | 76.0 |
| 450 | 232.0 | 8.0 | 13.4 | 52.0 | 71.5 |
| 500 | 260.0 | 7.5 | 12.8 | 49.0 | 65.0 |
| 550 | 287.8 | 7.8 | 12.3 | 43.5 | 60.0 |
| 600 | 316.0 | 7.7 | 11.8 | 39.0 | 53.0 |
| 700 | 371.0 | 8.2 | 11.55 | 35.5 | 54.5 |
| 800 | 427.0 | 8.7 | 11.9 | 37.5 | 54.0 |
| 850 | 455.0 | 7.13 | 9.9 | 27.5 | 41.0 |

TABLE V.—COPPER-ALUMINUM, DECEMBER 7, 1900.
ROLLED 1-INCH ROD.

*Analysis.*

| | Per cent. |
|---|---|
| Aluminum | 7.15 |
| Iron | 0.115 |
| Silicon | 0.027 |

*Tensile Tests.*

In tests marked * test bars were specially annealed in laboratory 2 hours at 600 degrees C.

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 60 | 15.5 | 4.26 | 23.86 | 91.0 | 72.9 |
| 400 | 204.0 | 4.93* | 20.94 | 81.0 | 58.2 |
| 500 | 260.0 | 4.31* | 17.95 | 49.0 | 43.7 |
| 600 | 316.0 | 4.93 | 14.56 | 25.0 | 30.4 |
| 800 | 427.0 | 4.93 | 10.02 | 12.0 | 16.0 |
| 900 | 482.0 | 2.84 | 6.25 | 13.0 | 25.4 |

TABLE VI.—COPPER-ZINC, JANUARY 16, 1900. EXTRUDED ROD.

Zinc about 40 per cent.

*Tensile Tests.*

In tests marked * test bars were annealed 2 hours at 600 degrees C. All other bars annealed, but temperature not recorded.

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 60 | 15.5 | 5.96* | 26.13 | 50.5 | 66.3 |
| 400 | 204.0 | 7.67* | 20.11 | 64.0 | 70.8 |
| 400 | 204.0 | 7.95 | 19.56 | 63.5 | 72.9 |
| 500 | 260.0 | 7.38 | 17.56 | 67.5 | 66.3 |
| 600 | 316.0 | 5.68 | 12.47 | 35.5 | 48.1 |
| 700 | 371.0 | 1.98 | 6.85 | 36.0 | 45.2 |
| 800 | 427.0 | 1.13 | 3.48 | 27.0 | 39.1 |

TABLE VII.—MILD STEEL, MARCH 23, 1900.
ROLLED 1-INCH ROD.

*Analysis.*

| | Per cent. |
|---|---|
| Silicon | 0.053 |
| Sulphur | 0.041 |
| Manganese | 0.401 |
| Nickel | 0.045 |
| Phosphorus | 0.138 |
| Carbon | Not determined. |

*Tensile Tests.*

In tests marked * test bars were specially annealed in laboratory at 600 degrees C.

| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 57* | 14 | 11.98 | 23.55 | 42.0 | 68.6 |
| 300 | 150 | 10.79 | 26.27 | 25.0 | 61.5 |
| 300 | 150 | 13.06 | 26.77 | 22.0 | 61.5 |
| 400 | 204 | 10.22 | 27.54 | 25.5 | 53.7 |
| 400 | 204 | 12.50 | 28.77 | 25.5 | 56.4 |
| 400 | 204 | 10.85 | 29.4 | 24.0 | 49.0 |
| 400* | 204 | 12.5 | 28.27 | 24.0 | 59.0 |
| 500 | 260 | 10.22 | 30.0 | 23.5 | 53.7 |
| 500 | 260 | 10.0 | 25.6 | 34.0 | 60.5 |
| 600 | 316 | 9.09 | 27.9 | 29.0 | 56.4 |
| 700 | 371 | 7.38 | 24.31 | 35.0 | 64.0 |
| 800 | 427 | 9.09 | 20.34 | 30.0 | 68.6 |

TABLE VIII.—WROUGHT IRON, MARCH 28, 1900.
ROLLED 1-INCH ROD.

| *Analysis.* | Per cent. |
|---|---|
| Silicon | 0.130 |
| Sulphur | nil |
| Manganese | nil |
| Nickel | nil |
| Phosphorus | 0.143 |
| Carbon | not determined. |

*Tensile Tests.*

In test marked * test bar was specially annealed in laboratory 2 hours at 600 degrees C.

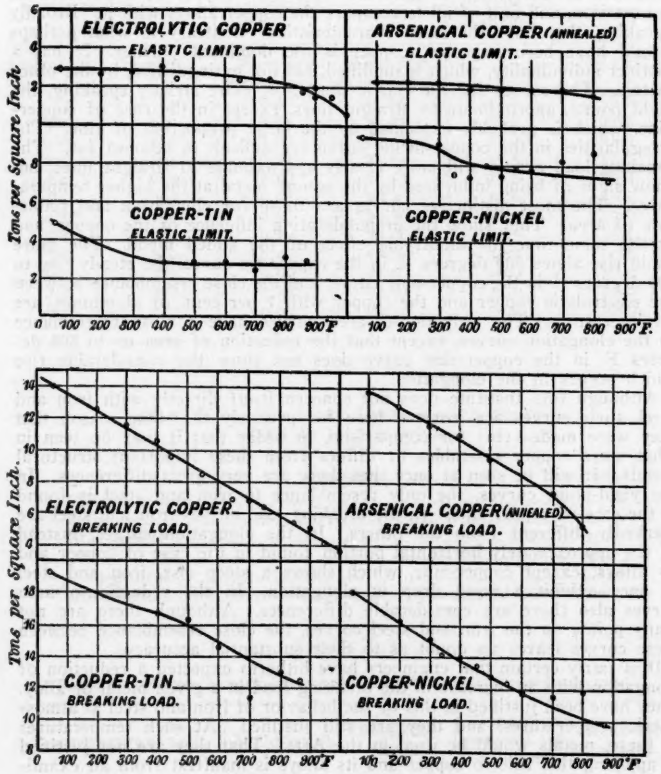
| Temperature of test. | | Yield-point. | Breaking load. | Elongation per cent. on 2 inches. | Reduction of area. |
|---|---|---|---|---|---|
| Degrees F. | Degrees C. | Tons per square inch. | Tons per square inch. | | Per cent. |
| 57 | 14 | 10.8 | 22.0 | 36.5 | 50.0 |
| 300 | 150 | 11.36 | 30.27 | 22.0 | 39.1 |
| 300 | 150 | 13.06 | 28.36 | 20.0 | 39.1 |
| 400 | 204 | 12.50 | 29.54 | 24.0 | 39.1 |
| 400* | 204 | 11.36 | 28.77 | 20.0 | 39.1 |
| 500 | 260 | 11.36 | 30.0 | 24.5 | 36.0 |
| 600 | 316 | 7.95 | 28.45 | 23.0 | 36.0 |
| 700 | 371 | 8.52 | 25 0 | 26.0 | 36.0 |
| 800 | 427 | 9.09 | 20.59 | 20.0 | 45.2 |

The foregoing results are shown graphically in the curves.

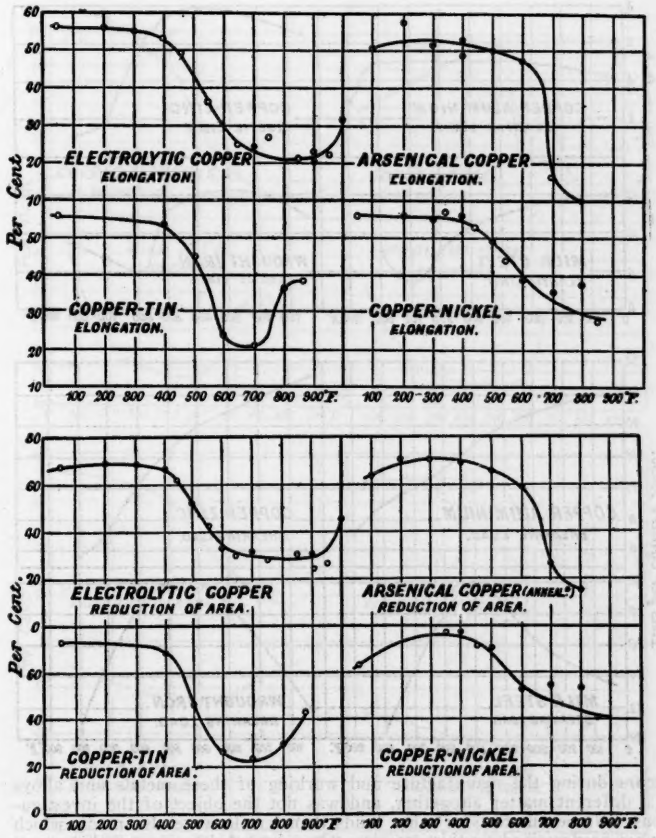
It will be convenient to take the electrolytic copper as a standard of comparison, and first of all to compare the copper alloys with it. Broadly speaking, they have the same characteristics. Contrary to what perhaps would have been expected, copper is the dominant partner. It has a distinct individuality, which is modified, but not extinguished, by the other metals. The curves for the elastic limits, or, more strictly speaking, the yield points, approximate to straight lines, except in the case of copper-zinc, which is probably explained by the large proportion of zinc. The irregularities in the copper-nickel curve are difficult to account for. The breaking-load curves still more closely approximate to straight lines, but show signs of being influenced by the second metal at the higher temperatures. The most distinctive curves are those for elongation and reduction of area. They show the preponderating influence of the copper, and at the same time the modifying effect of the added metal. The more rapid rise above 800 degrees F. in the copper-tin curve, the steady rise to 500 degrees F. in the copper-zinc curve, and the close resemblance between the electrolytic copper and the copper with 7 per cent. of aluminum, are all interesting. The reduction-of-area curves have a close resemblance to the elongation curves, except that the reduction of area up to 500 degrees F. in the copper-zinc curve does not show the considerable rise which occurs in the elongation.

Although this Institute does not concern itself directly with iron and steel, their curves are inserted here for precisely the same reason that they were made—viz: for comparison, in order that it may be seen in what ways copper resembles or differs from these important structural metals. It will be seen at once that there are very great differences. In the yield-point curves, the only resemblance to iron and steel is found in the case of copper-zinc. In the breaking-load curves iron and steel are markedly different from the others. In the elongation curves, instead of the approximately horizontal portion found in the case of copper and its alloys, except copper-zinc, which shows a steep rise, iron and steel at once exhibit a rapid drop in elongation. In the reduction-of-area curves also there are considerable differences. Although there are not many points on the iron and steel curves, the close resemblance between these curves leaves no doubt as to their substantial accuracy.

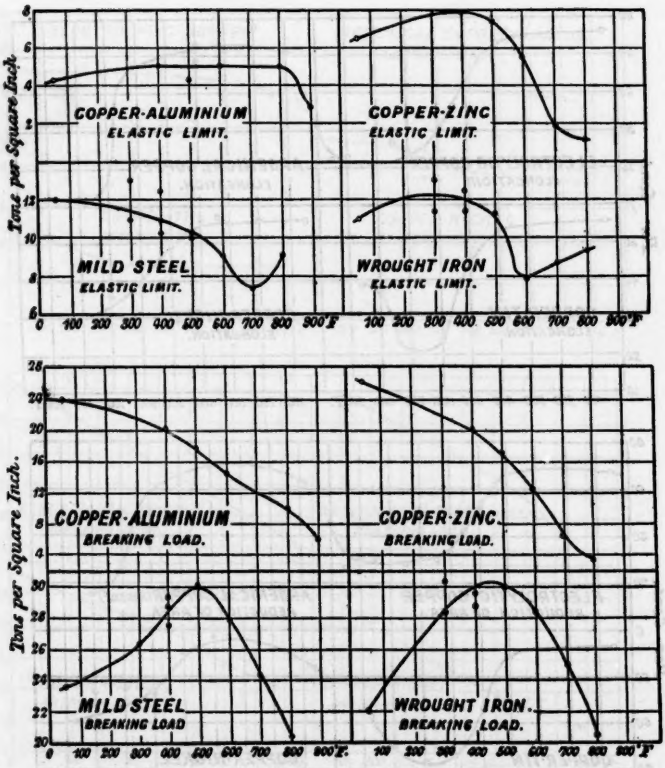
It is fairly certain that engineers have hitherto expected a reduction of elongation with an increase in the breaking load in a given metal or alloy. They have been justified in this by the behavior of iron and steel at atmospheric temperatures, and they are still justified. At such temperatures as these, metals would be used in the Arts. That they are not justified in applying this rule to copper and its alloys is manifest from an examination of the curves in this paper. It will be seen that, whilst the breaking loads are essentially straight-line curves, the elongation curves are contorted in perfectly definite ways, which, however, have no relation to the breaking loads. These are exceedingly interesting points and worthy of further study. It is somewhat remarkable that they have escaped observation and investigation so long. This is probably due to the fact that in such experiments as have been made, insufficiently pure metal has been used, the points observed have been taken too far apart, the test pieces have not been suitably annealed to make the tests comparable, and, not improbably, the carrying out of the tests has left something to be desired.




It is difficult to conceive that the definite and strongly-marked changes in direction in the elongation curves are not due to molecular disturbances. It is still more difficult to imagine these marked molecular disturbances leaving the breaking loads unaffected to a corresponding extent. Yet that is what takes place with copper and its alloys, and with iron and steel above a certain temperature.

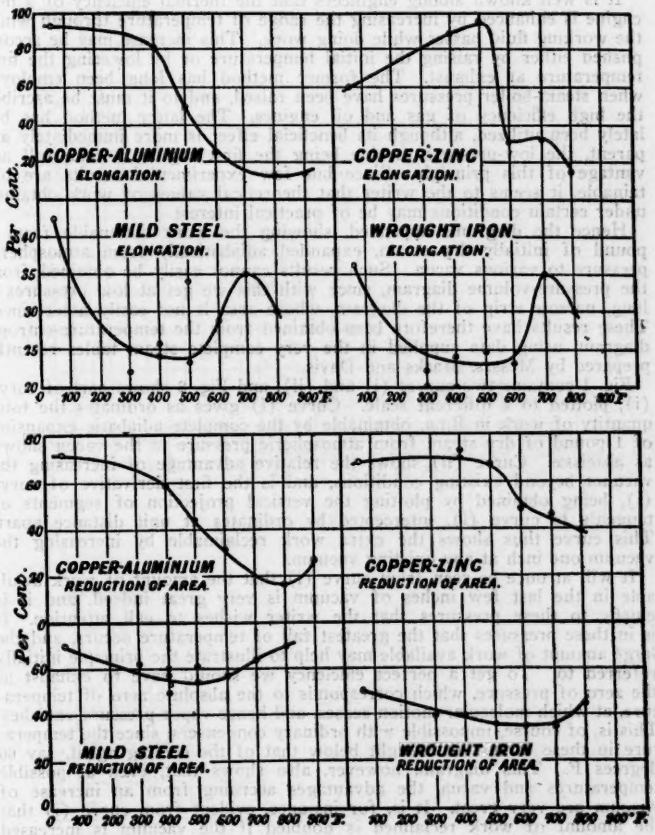



The range of temperatures given in this paper is restricted to such as might occur during the use of these metals and alloys in the Arts. What




occurs during the manufacture and working of these metals and alloys is a different matter altogether, and was not the object of the investigations. Numerous tests of other kinds than tensile were also made, which cannot be described on this occasion. It is intended to get out the curves for nickel, for which purpose rods were obtained some years ago.

The tests contained in this paper were carried out by different assistants, most recently by Mr. Licence and Mr. Baker. The author takes this




opportunity of expressing to all of them his gratitude for the zeal and efficiency which has invariably characterized their work.—"Engineering."

## ON THE IMPORTANCE OF MAINTAINING A HIGH VACUUM IN CONDENSER OF A STEAM TURBINE.

By S. L. HALL.

It is well known among engineers that the thermal efficiency of a heat engine is enhanced by increasing the range of temperature through which the working fluid passes while doing work. This increase may be accomplished either by raising the initial temperature or by lowering the final temperature at exhaust. The former method has long been employed when steam-boiler pressures have been raised, and to it must be ascribed the high efficiency of gas and oil engines. The latter method has but lately been utilized, although its beneficial effect is more immediately apparent, the low-pressure turbine being the first engine to take full advantage of this principle. Since but few experimental results are obtainable, it seems to the writer that theoretical values of work obtained under certain conditions may be of practical interest.

Hence the diagrams appended, showing the work obtainable from 1 pound of initially dry steam, expanded adiabatically from atmospheric pressure to various vacua. Such results cannot easily be obtained from the pressure-volume diagram, since with this we get at low pressures a long, narrow strip of the diagram, whose area is not easily ascertained. These results have therefore been obtained from the temperature-entropy diagram, using data supplied in the very complete steam tables recently prepared by Messrs. Marks and Davis.

Fig. 1 contains two curves (i) and (ii), and Fig. 2 shows part of curve (i), plotted to a different scale. Curve (i) gives as ordinates the total quantity of work in B.t.u. obtainable by the complete adiabatic expansion of 1 pound of dry steam from atmospheric pressure to the vacua shown as abscissæ. Curve (ii) shows the relative advantage of increasing the vacuum beyond existing conditions, and is the first derivative of curve (i), being obtained by plotting the vertical projection of segments of tangents to curve (i), intercepted by ordinates at unit distance apart. This curve thus shows the extra work reclaimable by increasing the vacuum one inch at any existing vacuum.

It will at once be seen from curve (i) that the amount of work available in the last few inches of vacuum is very great indeed, and it is chiefly to these pressures that the writer wishes to call attention. It is in these pressures that the greatest fall of temperature occurs, and the large amount of work available may help to illustrate the principle initially referred to. To get a perfect efficiency we should have to exhaust at the zero of pressure, which corresponds to the absolute zero of temperature, at which molecular motion ceases, and hence vapor pressure vanishes. This is, of course, impossible with ordinary condensers, since the temperature in these cannot be brought below that of the cooling agent, say 60 degrees F. This diagram, however, also shows that, even at possible temperatures and vacua, the advantages accruing from an increase of vacuum are very great. It is, for instance, evident from curve (i) that the amount of work reclaimed is doubled if the vacuum is increased from 23.8 inches to 29 inches, while from curve (ii) it may be seen that the importance of the vacuum at 29 inches is three times that at 26.9 inches, and that still lower pressures are even more desirable and important. Thus the losses arising from the use of condensers in which vacua of 23 or 24 inches are employed will be at once apparent, and the advantage of using the most recent improvements in the design of condensers and pumps for low-pressure turbine work will be evident.

These remarks, of course, apply only to the low-pressure turbine, and not to the reciprocating engine, in which the expansion is incomplete. The economy of using high vacua in the latter case is doubtful, since the

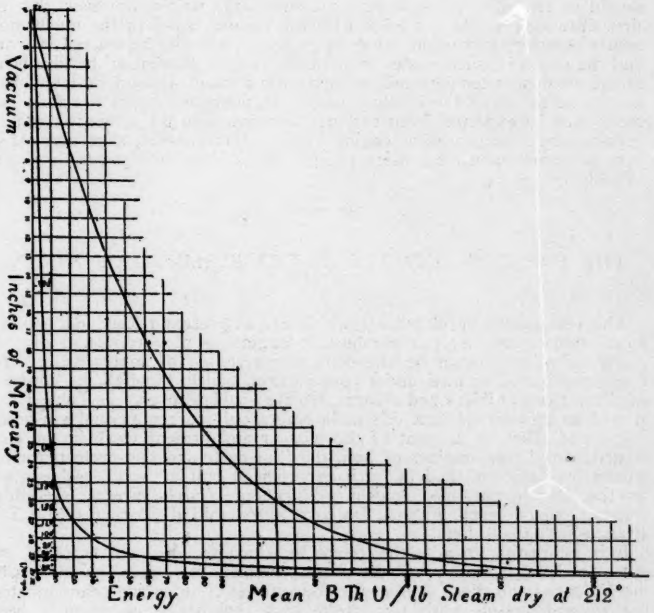



FIG. 1.

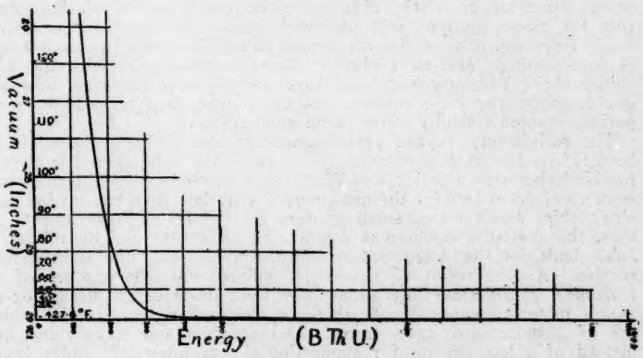



FIG. 2.

large amount of cooling water needed, and extra pump expense, more than neutralize the advantage of decreasing the back pressure another

pound or two per square inch. It seems to the writer, however, that the first drawback would not exist with the turbine, since in the much more nearly complete expansion, much more heat is usefully abstracted in work, and the steam is much wetter on admission to the condenser, besides being at the condenser temperature, so that only a small amount of latent heat has to be taken by the cooling water. It therefore seems probable that much may be expected from turbines working with the exhaust steam of high-pressure reciprocating engines, and converting the large amount of heat available during expansion to high vacua into useful work.—"Page's Weekly."

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## THE PRESENT ACTIVITY IN THE BUILDING OF MOTOR SHIPS.

The remarkable developments which are at present taking place in the application of the internal-combustion engine to the propulsion of sea-going ships are shown by the comprehensive list of motor vessels recently completed or now under construction, which, through the courtesy of the various builders and owners, we are enabled to give in Table I. It should be understood that this table refers only to commercial sea-going ships, and takes no account of the pioneer and present work in the construction of Diesel engines of both the four-cycle and two-cycle types for submarines accomplished by such experienced builders of Diesel engines as the Maschinenfabrik Augsburg-Nürnberg, Carels Frères, Gebrüder Sulzer, Fried. Krupp, Fabrica Italiana Automobili di Torino (F.I.A.T.), Franco Tosi, and others.

It is interesting to note the extent to which the different types of engines are being adopted. At the present moment, owing to the success of the *Vulcanus* and the *Selandia*, the four-stroke cycle engine predominates, but several vessels with two-stroke cycle engines have recently been completed, while a large number are under construction, as are also several vessels with Junker's tandem engines. Experience must be awaited before it is possible to form a definite opinion with regard to the comparative merits of the different types, but it seems not improbable that the two-cycle type will ultimately prove the superior for marine work, especially if a satisfactory design of double-acting two-cycle engine is forthcoming. Messrs. Blohm & Voss, in association with the Maschinenfabrik Augsburg-Nürnberg, have been at work for some time with double-acting two-cycle engines, but up to now they seem to have experienced some difficulty in obtaining good results.

The satisfactory voyage performances of the *Vulcanus* have already been referred to in these columns. As a result the builders of her engines, the Nederlandsche Fabriek van Werktuigen en Spoorweg-Materieel, have since received orders for the machinery for no less than ten further merchant ships and for two small cruisers for the Dutch Government. Of these the *Sembilan* has been in commission since 1911, and the oil carrier *Juno*, built for the Anglo-Saxon Petroleum Co., has just entered upon service. As the result of experience gained with the engines of the *Vulcanus*, considerable alterations have been made in the design of the *Juno's* motors, compared with those of the earlier vessel. The motors are of a more open type, turned steel columns and stays being used instead of a box casting for supporting the cylinders. Cylinder covers are dispensed with, the cylinders being cast with solid heads and the lower parts made detachable for the removal of the pistons. With this arrangement, the pistons can be removed in much less time, and there is no disturbance of the cam levers. A further alteration is the installation of a system of sight-drip lubrication, instead of forced lubrication. The

| | Type. | Name. | Completion | Owners. | Lgth. | Bdth. | Depth Mid. | Load Draft. | D.W. | Builders of Hull. | | Speed in Knots | No. of Propellers |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Oil Carrier | *Vulcanus* | 1910 | Anglo-Saxon Petroleum Co. Ltd., London. | 198'-0" | 37'-9" | 13'-2½" | 10'-2" | 1,180 | Nederlandsche Scheepbouw M., Amsterdam. | 1 | 8½ | 1 |
| 2 | Cargo Ship | *Sembilan* | 1911 | Koninklyke Paketvaart M., Amsterdam. | 152'-0" | 26'-0" | 9'-6" | 7'-0" | 250 | Do. | 2 | 8 | 1 |
| 3 | Oil Carrier | *Juno* | 1912 | Anglo-Saxon Petroleum Co. Ltd., London. | 258'-0" | 43'-0" | 19-10½ | 18'-6" | 2,675 | Do. | 3 | 10½ | 1 |
| 4 | Do. | *Emanuel Nobel* | 1912 | Soc. An. d'Armement, d'Industrie et de Commerce, Antwerp. | 375'-0" | 51'-0" | 29'-0" | 23'-0" | 6,230 | Do. | 4 | 11 | 2 |
| 5 | Do. | — | 1913 | Do. | 375'-0" | 51'-0" | 29'-0" | 23'-0" | 6,230 | Tyne Iron Shipbdg. Co. Ltd., Willington Quay. | 5 | 11 | 2 |
| 6 | Do. | *Eburra* | 1913 | Anglo-Saxon Petroleum Co. Ltd., London. | 346'-8" | 46'-6" | 27'-5" | 22'-0" | 5,050 | Nederlandsche Scheepbouw M., Amsterdam. | 6 | 10½ | 2 |
| 7 | Do. | *Artemis* | 1913 | Do. | 346'-8" | 46'-6" | 27'-5" | 22'-0" | 5,050 | Do. | 7 | 10½ | 2 |
| 8 | Do. | *Unio* | 1913 | Do. | 346'-8" | 46'-6" | 27'-5" | 22'-0" | 5,050 | Palmers Shipbdg. & Iron Co. Ltd., Hebburn. | 8 | 10½ | 2 |
| 9 | Do. | *Cassis* | 1913 | Do. | 346'-8" | 46'-6" | 27'-5" | 22'-0" | 5,050 | Caledon Shipbdg. & Eng. Co. Ltd., Dundee. | 9 | 10½ | 2 |
| 10 | Cargo Ship | *Loudon* | 1913 | Koninklyke Paketvaart M., Amsterdam. | 278'-0" | 41'-0" | 19'-0" | 15'-0" | 1,750 | Rykee Yard, Rotterdam. | 10 | 10½ | 1 |
| 11 | Do. | *Jules Henry* | 1913 | A Vimont et Cie, Marseilles. | 250'-0" | 41'-0" | 23'-5½" | 19'-0" | 2,500 | Wilton's Eng. & Slipway Co., Rotterdam. | 11 | 10½ | 2 |
| 12 | Lake Freighter | *Toiler* | 1910 | Canadian Lake Co., Montreal. | 248'-0" | 42'-6" | 19'-0" | 14'-0" | 2,600 | Swan, Hunter, & Wigham Richardson Ltd., Newcastle-on-Tyne. | 12 | 6½ | 2 |
| 13 | Do. | *Calgary* | 1912 | Swan, Hunter, & Wigham Richardson Ltd., Newcastle-on-Tyne. | 248'-0" | 42'-6" | 19'-0" | 14'-0" | 2,460 | W. Harkess & Son, Middlesbrough. | 13 | 7½ | 2 |
| 14 | Do. | — | 1913 | Montreal Transportation Co., Montreal. | 250'-0" | 42'-6" | 19'-0" | 14'-0" | 2,400 | Do. Swan, Hunter, & Wigham Richardson Ltd., Newcastle-on-Tyne. | 14 | 8 | 1 |
| 15 | Cargo Ship. | — | 1913 | — | 350'-0" | — | — | — | 5,600 | Do. | 15 | — | 2 |
| 16 | Oil Carrier | — | 1913 | Lane & Macandrew, London. | 310'-0" | 46'-0" | 27'-0" | — | 5,000 | Caledon Shipbdg. & Eng. Co. Ltd., Dundee. | 16 | 10½ | 2 |
| 17 | Cargo & Passenger | *Selandia* | 1912 | Akt. det Ostasiatiske Ko., Copenhagen. | 370'-0" | 53'-0" | 30'-0" | 25'-6" | 7,400 | Burmeister & Wain, Copenhagen. | 17 | 11½ | 2 |
| 18 | Do. | *Christian X.* | 1912 | Hamburg-Amerika Line, Hamburg. | 370'-0" | 53'-0" | 30'-0" | 23'-6" | 7,400 | Do. | 18 | 11½ | 2 |
| 19 | Cargo Ship | No. 285 | 1913 | A. B. Nordstjernan, Stockholm. | 362'-0" | 51'-3" | 34'-0" | 23'-0" | 6,500 | Do. | 19 | 10½ | 2 |
| 20 | Do. | 286 | 1913 | Do. | 362'-0" | 51'-3" | 34'-0" | 23'-0" | 6,500 | Do. | 20 | 10½ | 2 |
| 21 | Cargo & Passenger | 287 | 1913 | Akt. det Ostasiatiske Koa Copenhagen. | 410'-0 | 55'-0" | 38'-6" | — | 9,000 | Do. | 21 | 11½ | 2 |
| 22 | Do. | 288 | 1913 | Do. | 410'-0" | 55'-0" | 38'-6" | — | 9,000 | Do. | 22 | 11½ | 2 |
| 23 | Cargo Ship | 289 | 1913 | Foreign. | 362'-0" | 51'-3" | 34'-0" | 23'-0" | 6,500 | Do. | 23 | 10½ | 2 |
| 24 | Do. | 290 | 1913 | Do. | 362'-0" | 51'-3" | 34'-0" | 23'-0" | 6,500 | Do. | 24 | 10½ | 2 |
| 25 | Cargo & Passenger | *Jullandia* | 1912 | Akt. det Ostasiatiske Ko., Copenhagen. | 370'-0" | 53'-0" | 30'-0" | 23'-6" | 7,400 | Barclay, Curle & Co. Ltd., Glasgow. | 25 | 11½ | 2 |
| 26 | Cargo Ship | *Eavestone* | 1912 | Furness, Withy & Co., Ltd., West Hartlepool | 276'-0" | 40'-6" | — | — | 3,150 | Sir Rayiton Dixon & Co. Ltd., Middlesbrough. | 26 | 10 | 1 |
| 27 | Auxiliary Barque | *La France* | 1912 | Prentout-Leblond Leroux et Cie., Rouen. | 390'-5" | 57'-3" | 28'-3" | 23'-8" | 6,500 | Chantiers de la Gironde, Bordeaux. | 27 | 10 | 2 |
| 28 | Lake Freighter | *Fordonian* | 1912 | Mr. J. W. Norcross, Port Colborne, Ont. | 250'-0" | 42'-6" | 16'-10" | — | 4,000 | Clyde Shipbdg. & Eng. Co., Port-Glasgow. | 28 | 9 | 1 |
| 29 | Cargo Ship | *Rolandseck* | 1912 | Deutsche D. G. Hansa, Bremen. | 260'-0" | 40'-0" | 28'-0" | 18'-0" | 2,600 | J. C. Tecklenborg, A. G., Geestemunde. | 29 | 11½ | 1 |
| 30 | Oil Carrier [ger | — | 1912 | Deutsch-Amerikanische Pet. G., Hamburg. | 404'-0" | 52'-5½" | 29'-6" | 23'-0" | 7,500 | Reiherstieg Schiffswerfte & Maschinenfabrik, Hambg. | 30 | — | 1 |
| 31 | Cargo & Passen- | *Romagna* | 1911 | Romagnola S.S. Co. | 175'-5" | 26'-3" | 12'-7" | 11'-6" | — | Cantieri Riuniti, Ancona. | 31 | 12½ | 2 |
| 32 | Cargo Ship | *Monte Penedo* | 1912 | Hamburg-Sudamerikanische D.G., Hamb. | 350'-0" | 50'-0" | 27'-0" | — | 6,500 | Howaldtswerke, Kiel. | 32 | 10½ | 2 |
| 33 | Do. | *Primus* | 1912 | Hamburg-Amerika Line, Hamburg. | 353'-0" | 50'-0" | 26'-0" | 22'-6" | 6,000 | A. G. Weser, Bremen. | 33 | 10 | 2 |
| 34 | Oil Carrier | No. 266 | 1913 | Steaua Romana, Bukarest. | 295'-3" | 44'-0" | 26'-3" | 21'-4" | 4,100 | J. Frerichs & Co A.G., Osterholz-Scharmbeck. | 34 | 10 | 2 |
| 35 | Do. | 269 | 1913 | Steaua Romana, Bukarest. | 385'-0" | 51'-6" | 30'-3" | 24'-3" | 7,200 | Do. | 35 | 11 | 2 |
| 36 | Do. | 186 | 1913 | Deutsch-Amerikanische Pet. G., Hamburg. | 400'-0" | 53'-0" | 32'-4" | 25'-3" | 7,700 | Fried. Krupp A.G., Germaniawerft, Kiel. | 36 | 11 | 2 |
| 37 | Do. | 187 | 1913 | Do. | 400'-0" | 53'-0" | 32'-4" | 25'-3" | 7,700 | Do. | 37 | 11 | 2 |
| 38 | Do. | 188 | 1913 | Do. | 525'-0" | 66'-3" | 33'-6" | — | 14,000 | Do. | 38 | 11 | 2 |
| 39 | Cargo Ship | — | 1913 | Blohm & Voss, Hamburg | — | — | — | — | — | Blohm & Voss, Hamburg. | 39 | — | 2 |
| 40 | Do. | — | 1913 | Hamburg-Amerika Linie | — | — | — | — | — | Do. | 40 | — | 2 |
| 41 | Auxiliary Barque | *Quevilly* | 1911 | Prentout-Leblond Leroux et Cie, Rouen. | 303'-6" | 45'-3" | 26'-7" | — | 3,900 | Chantiers de Normandie, Rouen. | 41 | 6½ | 2 |
| 42 | Oil Carrier | — | 1912 | Standard Oil Co. | 200'-0" | 35'-0" | 15'-6" | 14'-0" | 1,500 | Staten Island Shipbdg. Co. | 42 | — | 1 |
| 43 | Cargo Ship | — | 1913 | Cantieri Officine Savoia. | — | — | — | — | — | Cantieri Savoia. | 43 | — | 1 |
| 44 | Do. | — | 1913 | — | — | — | — | — | — | Soc. An. John Cockerill, Seraing. | 44 | — | 2 |
| 45 | Oil Carrier | *Delo* | 1908 | Merkulew Bros. | 355'-0" | 46'-0" | 25'-0" | 14'-0" | 4,100 | Kolomnaer Maschinenfabrik A.G., Golutwin, Russia. | 45 | 9½ | 2 |
| 46 | Do. | *Emanuel Nobel* | 1909 | Nobel Bros., St. Petersbg | 380'-0" | 46'-0" | 25'-0" | 16'-6" | 4,600 | Do. | 46 | 10 | 2 |
| 47 | Do. | *K. W. Hagelin* | 1910 | Do. | 380'-0" | 46'-0" | 25'-0" | 16'-6" | 4,600 | Do. | 47 | 10 | 2 |
| 48 | Do. | *Galileus* | 1911 | Do. | 270'-0" | 33'-4" | 20'-0" | 15'-6" | 2,000 | Do. | 48 | 11 | 2 |
| 49 | Do. | *Zoroaster* | 1911 | Do. | 270'-0" | 33'-4" | 20'-0" | 15'-6" | 2,000 | Do. | 49 | 11 | 2 |
| 50 | Do. | — | 1913 | British Admiralty. | — | — | — | — | 1,000 | Devonport Dockyard. | 50 | — | — |
| 51 | Do. | — | 1913 | Do. | — | — | — | — | 1,000 | Do. | 51 | — | — |

| Speed in Knots | No. of Pro-pellers | Builders of Engines. | Type of Engines. | Total B.H.P | No. of Eng. | Cylinders. No. per Eng. | Diam. | Stroke | Revs. per Min. |
|---|---|---|---|---|---|---|---|---|---|
| 8½ | 1 | Nederlandsche Fabrik van Werktingen en Spoorweg Materieel, Amsterdam | Werkspoor four-stroke cycle, single-acting, reversible. | 500 | 1 | 6 | 15½" | 23½" | 180 |
| 8 | 1 | Do. | Do. | 200 | 1 | 3 | 15½" | 19½" | 200 |
| 10½ | 1 | Do. | Do. | 1,100 | 1 | 6 | 22" | 39½" | 135 |
| 11 | 2 | Do. | Do. | 2,200 | 2 | 6 | 22" | 39½" | 135 |
| 11 | 2 | Do. | Do. | 2,200 | 2 | 6 | 22" | 39½" | 135 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 6 | 20½" | 35½" | 135 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 6 | 20½" | 35½" | 135 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 6 | 20½" | 35½" | 135 |
| 10½ | 1 | Do. | Do. | 1,100 | 1 | 6 | 22" | 39½" | 135 |
| 10½ | 2 | Do. | Do. | 1,000 | 2 | — | — | — | — |
| 6½ | 2 | A. B. Diesels Motorer, Stockholm. | Marine "Polar," two-stroke cycle, single-acting, reversible. | 360 | 2 | 4 | 9½" | 14½" | 260 |
| 7½ | 2 | Do. | Do. | 520 | 2 | 4 | 11½" | 16½" | 250 |
| 8 | 1 | — | High-speed non-reversible Diesel engines, with Mavor electric transmission gear. | 600 | 2 | 6 | 12" | 13½" | 400 E. 80 Pro. |
| — | 2 | Swan, Hunter, & Wigham Richardson, Ltd., N'castle-on-T. | Two-stroke cycle, single-acting, reversible. | — | 2 | 4 | — | — | — |
| 10½ | 2 | A. B. Diesels Motorer, Stockholm. | Marine "Polar," two-stroke cycle, single-acting, reversible. | 2,000 | 2 | 6 | 17½" | 21½" | 165 |
| 11½ | 2 | Burmeister & Wain, Copenhagen | Four-stroke cycle, single-acting, reversible. | 2,100 | 2 | 8 | 20½" | 28½" | 140 |
| 11½ | 2 | Do. | Do. | 2,100 | 2 | 8 | 20½" | 28½" | 140 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 8 | 19½" | 26" | 140 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 8 | 19½" | 26" | 140 |
| 11½ | 2 | Do. | Do. | 2,500 | 2 | 8 | — | — | 140 |
| 11½ | 2 | Do. | Do. | 2,500 | 2 | 8 | — | — | 140 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 8 | 19½" | 26" | 140 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 8 | 19½" | 26" | 140 |
| 11½ | 2 | Barclay, Curle & Co. Ltd., Glasgow. | Do. (Burmeister & Wain design). | 2,100 | 2 | 8 | 20½" | 28½" | 140 |
| 10 | 1 | Richardsons, Westgarth & Co. Ltd., Middlesbrough. | Carels Frères two-stroke cycle, single-acting, reversible. | 800 | 1 | 4 | 20" | 36" | 95 |
| 10 | 2 | Schneider et Cie. | Do. | 1,800 | 2 | 4 | 17½" | 22" | — |
| 9 | 1 | Clyde Shipbuilding & Eng. Co., Port-Glasgow. | Do. | 800 | 1 | 4 | 15½" | 35½" | — |
| 11½ | 1 | Carels Frères, Ghent. | Do. | 1,500 | 1 | 6 | 20" | 36½" | 130 |
| — | 1 | Do. | Do. | 1,800 | 1 | 6 | 23½" | 43½" | 100 |
| 12½ | 2 | Gebr. Sulzer, Winterthur. | Two-str. cycle, single-actg., reversible. | 800 | 2 | 4 | 12½" | 18½" | 250 |
| 10½ | 2 | Do. | Do. | 1,700 | 2 | 4 | 18½" | 26½" | 160 |
| 10 | 2 | A. G. Weser, Bremen. | Two-stroke cycle, Junkers tandem arrangement, reversible. | 1,600 | 2 | 3 | 15½" | 2-15½" | 120 |
| 10 | 2 | J. Frerichs & Co. A.G. | Do. | 1,300 | 2 | 2 | 15½" | 2-17½" | 180 |
| 11 | 2 | Do. | Do. | 2,400 | 2 | 4 | 17½" | 2-20½" | 150 |
| 11 | 2 | Fried. Krupp A.G. | Two-stroke cycle, single-acting, reversible. | 2,300 | 2 | 6 | 18½" | 31½" | 140 |
| 11 | 2 | Do. | Do. | 2,300 | 2 | 6 | 18½" | 31½" | 140 |
| 11 | 2 | Do. | Do. | 3,500 | 2 | 6 | 22½" | 39½" | 125 |
| — | 2 | Blohm & Voss, and Maschinenfabrik Augsburg-Nürnberg A.G. | M.A.N. two-stroke cycle, double-acting, reversible. | 1,700 | 2 | 3 | 18½" | 25½" | 120 |
| — | 2 | Blohm & Voss. | Do. | 3,000 | 2 | 3 | — | — | — |
| 6½ | 2 | Chantiers Augustin Normand, Havre. | M.A.N. two-stroke cycle, single-acting reversible. | 600 | 2 | 6 | — | — | 300 |
| — | 1 | New London Co., Groton, T.S.A. | Do. | 300 | 1 | 6 | 9" | 13½" | 300 |
| — | 1 | — | Two-stroke cycle, single-act. reversible | 300 | 1 | 4 | 9½" | 15½" | — |
| — | 2 | Soc. An. John Cockerill. | Do. | 2,000 | 2 | 6 | — | — | 300 |
| 9½ | 2 | Kolomnaer Maschinenfabrik A.G. | Four-stroke cycle, single-acting, reversal by Von Koreiwo Clutch. | 1,000 | 2 | 4 | 17½" | 26½" | 200 |
| 10 | 2 | Do. | Do. | 1,200 | 2 | 4 | 19½" | 29" | 150 |
| 10 | 2 | Do. | Do. | 1,200 | 2 | 4 | 19½" | 29" | 150 |
| 11 | 2 | Do. | Four-stroke cycle, sgle-act., reversible | 1,000 | 2 | 4 | 17½" | 26½" | 200 |
| 11 | 2 | Do. | Do. | 1,000 | 2 | 4 | 17½" | 26½" | 200 |
| — | — | Vickers Ltd., Barrow. | — | — | — | — | — | — | — |
| — | — | J. S. White & Co., East Cowes. | — | — | — | — | — | — | — |

building of "Werkspoor" engines in this country is to be taken up by the North Eastern Marine Engineering Co., of Wallsend.

The *Selandia* has demonstrated, by her successful voyage of 20,000 knots to the Far East and back, the reliability of the Diesel engine at sea and the fuel economy to be obtained by the use of motor engines. On her outward voyage a speed of 11½ knots was maintained on a fuel consumption of about 10 tons per day, and on the homeward voyage she averaged 10 knots with 8½ tons per day. Her builders, Messrs. Burmeister & Wain, in addition to completing the sister ship *Fionia* for the same owners (now sold to the Hamburg-Amerika Line, and renamed *Christian X*), have orders for at least six other motor ships, and, at the time of going to press, are stated to have secured still further contracts for motor vessels. The *Jutlandia*, built by Messrs. Barclay, Curle & Co. on the Clyde, and installed with engines of Burmeister & Wain design, has also done well on her trials and is now at sea. Negotiations are now in progress for the building of new engine works in the Clyde district to construct the Burmeister & Wain type of engine.

Before leaving the question of the four-cycle engine as applied to large commercial vessels, mention must be made of the great work accomplished in Russia in this respect by the Kolumnaer Maschinenfabrik, of Golutwin, and the Ludwig Nobel Maschinenfabrik, of St. Petersburg, which is perhaps not generally recognized in this country. In Table I only those Russian vessels navigating the Caspian Sea are included, but a number of large vessels have also been built for service on the River Volga. The Kolumnaer Co. alone have completed or have under construction the engines for about forty such craft, with horsepowers from 200 to 1,200, while Messrs. Nobel have also installed a large number.

The pioneer sea-going motor ships of any considerable size installed with two-cycle engines were the *Romagna* and the *Toiler*. The former vessel, whose machinery was supplied by Messrs. Gebrüder Sulzer, was unfortunately lost at sea soon after completion, through no fault of the engines. Messrs. Sulzer have since completed the two-cycle engines for the *Monte Penedo*, which are described elsewhere in the present issue. The *Toiler*, which was built by Messrs. Swan, Hunter & Wigham Richardson, Ltd., after crossing the Atlantic under her own power, is now successfully employed on the Canadian Lakes. Her engines were supplied by the Aktiebolaget Diesels Motorer, who have had great experience in building two-cycle Diesel engines for smaller craft, and have now in hand two engines of 1,000 H.P. for a vessel building at Dundee. A similar ship to the *Toiler*, but with more powerful engines, also by the Aktiebolaget Diesels Motorer, has recently been completed by Messrs. Swan, Hunter & Wigham Richardson. The latter firm are also building at their Neptune Works a cargo vessel, 350 feet long and of about 5,600 tons deadweight, to be fitted with Diesel oil engines driving twin screws. The engines also are to be built at the Neptune Works, and will be of the reversible single-acting two-cycle type. Each engine will have four working cylinders, four scavenging cylinders, two air compressors, one circulating pump, and one bilge pump. These builders are also completing at their Neptune Works a marine Diesel engine of 400 B.H.P. for experimental purposes, and are largely rebuilding their engine works with a view to undertaking the construction of motor engines on a large scale.

The design of two-cycle engine which is finding most adoption is that developed by Messrs. Carels Frères, of Ghent. The motor ship *Eavestone*, built for Messrs. Furness, Withy & Co., and equipped with a Carels two-cycle engine, is now in service and has made several voyages. The results are said to be satisfactory, but no definite information has yet been published with regard to her performances, and the same may be said of the auxiliary barque *France*, which also has Carels engines. Two other vessels, the *Fordonian*, built by the Clyde Shipbuilding and Engi-

neering Co., and the *Rolandseck*, by Messrs. Tecklenborg, have been launched, and the former is now at sea. A fifth vessel building at the Reiherstieg Yard in Hamburg will have the largest power yet installed on a single shaft. Messrs. Carels Frères and the Diesel Engine Co. of London, have recently amalgamated in the formation of the Consolidated Diesel Engine Manufacturers, Ltd., and are now laying down extensive works at Ipswich for the construction of Diesel engines.

Another important firm who have taken up the construction of two-cycle engines on a large scale are Messrs. Fried. Krupp A.G., of Kiel, who have in hand three oil carriers to be propelled by oil engines of this type. The construction of the Junkers engine has been taken up by the A. G. Weser, of Bremen, and J. Frerichs & Co. A.G., of Osterholz-Scharmbeck. The former are building one large vessel, and the latter firm have two ships to be propelled by this type of motor. Messrs. Frerichs have also installed Junkers engines in two large deep-sea fishing boats, and have fitted two-cycle Diesel engines in a large number of such craft.

Several builders of marine engines are preparing to build Diesel marine engines in the event of their coming into general use. Messrs. William Doxford & Sons, of Sunderland, have constructed a single-cylinder two-cycle engine of their own design for experimental purposes. The cylinder of this engine has a diameter of 19½ inches and a stroke of 37 inches. The engine is of the open crosshead type, and has the usual scavenging valves, while the pistons, connecting rods, and all main parts are exposed and readily accessible. Another firm, Messrs. Vickers, Ltd., of Barrow, are busy with the construction of an experimental single-cylinder 1,000-H.P. engine of their own design and intended for naval work.

So far, attention has been confined in the present article to reversible Diesel marine engines. Another line on which development is proceeding is that of the employment of non-reversible high-speed Diesel engines in conjunction with some form of speed-reduction gearing. Such devices were employed for reversing only in the earlier Russian Diesel-engined boats, but were abandoned when the directly-reversible motor came into use. In association with speed-reduction gearing, however, there appear to be possibilities of economy for the high-speed non-reversible engine compared with the slow-speed directly-driving reversible engine, and two interesting vessels are now under construction in which this matter will be tested. The first is a Canadian Lake boat 250 feet long, 42 feet 6 inches broad, and 19 feet deep, which is to be fitted with the Mavor system of electrical-transmission gear, and is being built by Messrs. Swan, Hunter & Wigham Richardson. This vessel will have two high-speed non-reversible Diesel engines of a combined B.H.P. of 600, driving two sets of alternators which supply current to an electric motor driving a single screw. The engines will run at about 400 revolutions per minute, and the propeller at about 80 revolutions per minute. One great advantage of the system is that the engines can be controlled from the bridge. The other vessel is a Congo river boat, which is being built by Messrs. Cockerill, in Belgium, and is to have two high-speed non-reversible Diesel engines of about 640 H.P. each, driving twin propellers through Föttinger's hydraulic transformers.

We have endeavored, as far as possible in a short article, to sum up the present position with regard to motor sea-going merchant ships, but progress is taking place so quickly in the application of the Diesel engine to marine work that doubtless some omissions have occurred. This is particularly the case with regard to Government vessels, over which a certain amount of secrecy is maintained. It is, however, generally understood that the Admiralty have under construction two small Diesel-engined oil carriers at the Devonport Dockyard, one to be engined by Messrs.

Vickers, and the other by Messrs. J. Samuel White & Co., of East Cowes, while tenders for a larger oil carrier are under consideration.—"The Shipbuilder."

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## THE *SELANDIA'S* CONSUMPTION OF OIL.

As the consumption of oil is a very important feature in connection with the motor vessel, the following particulars of the *Selandia's* consumption of this fuel may be of interest. The subjoined table gives some of the worst and some of the best days, and the average speed for the different distances on the outward and the homeward voyage. The weather, especially on the homeward voyage, was anything but favorable, yet oil consumption and speed remained fairly stable; and it is interesting to note that the oil consumption has gone down almost 2 tons per day from the commencement of the trip to its finish. A steamer would have used three to four times as much coal.

*Outward Voyage.*

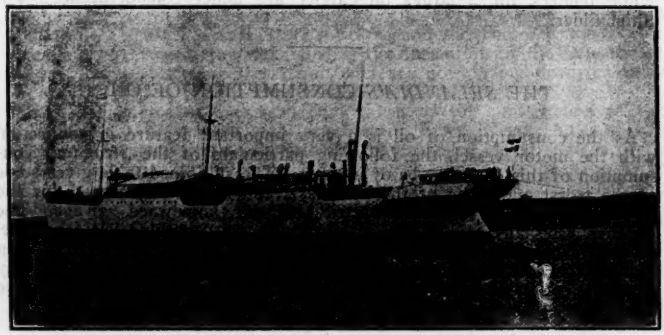
| | Consumption of oil per day, in tons. | Speed. Knots. |
|---|---|---|
| March 11, 1912: The Atlantic; heavy sea, vessel works hard, taking much water over | 9.8 | 9.2 |
| March 13, 1912: The Atlantic; vessel rolls, and takes much water over | 10.2 | 10.4 |
| March 14, 1912: The Atlantic; no special weather report | 10.0 | 11.5 |
| March 22, 1912: Mediterranean; no special weather report | 10.1 | 12.7 |
| March 25 to April 6, 1912: Suez to Colombo; no special weather report, average | 10.0 | 11.5 |
| April 9, 1912: no special weather report, average | 9.6 | 12.4 |
| April 8 to 11, 1912: Colombo-Penang; no special weather report, average | 9.6 | 11.5 |
| April 13, 1912: Penang-Bangkok; no special weather report, average | 9.6 | 11.7 |

*Homeward Voyage.*

| | | |
|---|---|---|
| May 11 to 16, 1912: Penang-Colombo; heavy, contrary sea, average | 8.8 | 9.6 |
| May 18 to June 1, 1912: Colombo-Suez; three days' heavy contrary sea, vessel working hard, taking much water over | 8.7 | 10.0 |
| June 2 to 8, 1912: Port Said-Genoa; no weather report, average | 8.7 | 10.4 |
| June 10 to 19, 1912: Genoa-Dover; heavy contrary sea and weather | 8.4 | 9.6 |
| June 19 to 22, 1912: Dover-Aarhus; no weather report | 8.0 | 10.3 |

The success of the *Selandia* and her sister ship, sold to the Hamburg-Amerika Line, has brought their builders—Messrs. Burmeister & Wain—several orders, and they now have orders for nine motor vessels, of an aggregate tonnage of 40,000 tons gross, whilst their orders for steamers is confined to one—a screw boat.—"Engineering."

## A 10,000-TON MOTOR-DRIVEN SHIP.



THE FIRST MOTOR-DRIVEN FREIGHT AND PASSENGER SHIP TO ARRIVE AT NEW YORK.

Twin motors of 2,500 combined horsepower drove this ship from Hamburg to Havana, 4,627 miles, at a speed of 11.1 knots and with a consumption of 0.34 pound of oil per horsepower per hour. Note the absence of smokestacks.

THE TWIN MOTORS OF THE "CHRISTIAN X," EACH OF 1,250 HORSEPOWER.

A steam plant would reduce the cargo capacity of this ship by 20 per cent. The saving due to the oil motors is $13,000 every 100 days.

The arrival of the *Christian X*, the first large ocean-going, motor-driven ship, at New York, marks another important step in the development of transatlantic travel, an advancement of greater significance than any that has been made since a steamship first crossed the Western Ocean. Had the *Christian X* been a smaller ship of slower speed, and driven by motors of moderate power, the successful passage just accomplished would not be so full of meaning. As matters stand, no one can deny that the era of the motor-driven ocean liner has now been fairly well launched, and the question of applying the oil engine to the propulsion of the fastest and largest of transatlantic liners is henceforth merely one of mechanical development; the 25-knot, 50,000-ton ship is clearly within sight.

The *Christian X*, which was built by Burmeister & Wain, Ltd., of Copenhagen, and is owned by the Hamburg-American Company, is a freight and passenger vessel, whose length is 370 feet, beam 53 feet, and maximum draught 24 feet, at which her displacement is 10,200 tons, and her dead-weight carrying capacity 5,500 tons. She is driven by twin Diesel engines of a combined maximum indicated horsepower of 2,500.

Each engine contains eight cylinders, 20⅞ inches in diameter, and 28¾ inches stroke. In each cylinder head are four valves, respectively for the air admission, fuel admission, the exhaust, and for starting by compressed air. The charge is ignited, on the well-known Diesel system, by the heat of compression, the air being compressed to 475 pounds and corresponding temperature of 650 degrees C. In starting the engines, compressed air is drawn for the purpose from two reservoirs at a pressure of 300 pounds to the square inch. The compressed air is utilized for the first three or four strokes, when the air supply is cut out and the oil supply cut in.

There are two auxiliary engines, one on the outboard side of each main engine. Each is a four-cylinder, Diesel motor of 250 horsepower and each is connected to a dynamo for furnishing electric light and power, and to a four-stage air compressor, which is used for supplying air at 300 pounds to the two reservoirs used in starting the main engines. The air for atomizing the oil is further compressed to 900 pounds to the square inch.

The main engines are controlled from the starting platform by means of a lever and a wheel, the first regulating the oil supply, the second the reversing. The reversing gear and the general mechanism for operation of the valves are among the most successful features of the design. We are informed by the chief engineer that, during the tests, with the engines running full speed ahead, the port engine was changed to full speed astern in 10 seconds' time, a remarkable result in an engine of 1,260 horsepower.

That the first Atlantic crossing by a motor-driven ship of large size has been achieved with perfect success is shown by the following facts: The *Christian X*, with a full load aboard, left Hamburg July 23d, and reached Havana August 9th, having covered 4,627 miles, without a stop, at an average speed of 11.1 knots. The chief engineer informs us that the weather was decidedly bad from the 28th of July to the 5th of August, with winds of from 7 to 8 strength, and a sea which caused the ship to pitch and roll heavily and take much water on deck. The propellers were frequently out of the water; but the Aspinall regulator, which acts directly on the oil feed, gave excellent results, as may be judged from the fact that with revolutions of from 140 to 145, there was a gain of only three revolutions while the propellers were out of the water.

The ship carries 1,000 tons of oil in her double bottom. She left Hamburg with a supply of 470 tons and reached Havana with 290 tons, having run at full speed with a fuel consumption of about 10 tons per day. The engines, being new, were run at only 135 revolutions per minute for the first few days; later this was raised to 142 revolutions. The best day's run was 329 miles, at an average of 13.3 knots per hour. It is expected

that when the engines are shaken down the ship will be good for an average speed of 13 knots.

The fuel consumption for the main engines and all auxiliaries worked out at 0.34 pound per indicated horsepower per hour. Outside of this economy in fuel consumption, the adoption of the oil motor for transatlantic ships such as this presents many and great advantages. In the first place, in a ship of this size 1,000 tons additional of freight can be carried, due to the fact that the space that would have been given up to the boiler plant is now available for cargo. Secondly, there is a reduction of fifteen men in the crew, namely, six stokers, six trimmers and three oilers. The present engine-room force consists of seven engineers, one storekeeper and one electrical engineer. The engines ran throughout the trip to Havana with great regularity and without a single stop during the whole seventeen days consumed on the voyage.

An estimate of the economy in this ship over one driven by steam engines shows that in one hundred days of operation there would be a saving of about $13,000, or say, about $50,000 in a single year.—"Scientific American."

## GERMAN MOTOR PROGRESS.

We believe we are correct in saying that there are only two German owned motor ships of any size afloat at the present moment. One, the *Christian X*, of which both the hull and engines were built in Denmark, was purchased by the Hamburg-American Company; the other, the *Monte Penedo*, had her hull built in Germany and her engines in Switzerland. We believe we are equally correct in saying that there is not a single large ship afloat of which both hull and engines have been built in Germany, ignoring such craft as submarines, etc. There are, on the other hand, three fairly large motor ships, the *Jutlandia*, the *Fordonian* and the *Eavestone*, of which both hull and engines have been built in Great Britain and which are now on active service. A bare statement of these facts alone might lead to the supposition that we in this country are ahead of our Continental friends in the building of motors for marine work. We think, however, that a more intimate knowledge of all that has actually been done in the respective countries would lead to quite a different conclusion as to the actual relative state of progress. As far as work under construction in this country is concerned, apart from the three ships already mentioned, as being actually in commission, the largest engine yet commenced is a 6,000 horsepower six-cylinder two-cycle single-acting engine which Vickers have on order for a British cruiser. Thornycrofts are building a six-cylinder two-cycle 1,000 horsepower engine for a destroyer, and Doxfords and Workman Clarks are experimenting on single-cylinder engines of their own design. Of these the 6,000-H.P. engine is at present very much in the air, having only reached the stage of single-cylinder experiment, and the others have not yet reached the trial-trip period. Apart from the thorough investigation of Continental practice which the above-mentioned and other firms have undertaken, and the licenses which they have taken out, this sums up fairly accurately the position over here, although Vickers have undoubtedly accumulated a lot of valuable information from their experience with the submarine engines already built by them.

A good idea of what has already been done in Germany may be obtained from the articles, of which we print the first today, recording a tour of inspection of the big German works, and it will be seen that the amount of progress there is not only greater but is based upon a great deal more solid foundation. There we found that Krupps, who have been studying the Diesel-engine problem since 1899, have completed and tried

one six-cylinder engine of 1,150 brake horsepower, a second being well under way, while another pair of 1,750 brake horsepower each already have their framework partly erected. There are, besides, many smaller sizes. Above all, some three years ago they built a single-cylinder double-acting engine of 2,000 brake horsepower, and put it through an exhaustive series of experiments which proved so satisfactory that the firm is now ready to accept an order under guarantee for a set of double-acting engines of 8,000 brake horsepower on a single shaft. Blohm & Voss have had sixteen months of constant experimenting with a three-cylinder double-acting engine of 1,000 brake horsepower, which has given them sufficient information to commence with perfect confidence the construction of other sets of the same power embodying the results of their experience; they are also building a single-acting set of 1,350 brake horsepower. The M.A.N., whose total experience goes back over fifteen years, has also had fifteen months on a double-acting engine of 1,000 brake horsepower, and considerable experience, the amount of which we can only guess, with what has always been reported to be a three-cylinder double-acting engine of 6,000 brake horsepower. From all we can learn, builders over here have entirely fought shy of even experimenting with the double-acting engine, which at the moment appears to be the only motor in which the maximum horsepowers necessary for naval purposes can be obtained. Further down the scale, if we may be allowed to put it so, the Reiherstieg Company has built an engine of 1,800 brake horsepower; Tecklenborg one of 1,500 brake horsepower; Frerichs are building a pair of 1,200 brake horsepower each, after good experience on smaller powers, and the Weser Yard is building two of 800 brake horsepower each, as well as some smaller sizes. The fundamental difference appears to us to be that whereas with the single exception of Vickers Limited, the three marine engines built in this country and actually at work are the first motors constructed by their respective builders—and that under the instruction of their licensors—apart from a single-cylinder experimental engine, which in one case was run for a few months while the other engine was actually under construction, some, at all events, of the German firms have the inestimable advantage of twelve to fifteen years' experience of building Diesel engines behind them, so that the construction of an engine of 1,500 horsepower has nothing experimental about it. As we have stated before, the art of constructing big Diesel engines cannot be learned in a day. Thus the Germans are apparently ahead of us in the accumulation of experience and experimental data. In three other directions we think, too, that they have a very great lead. They have obtained a greater horsepower per cylinder; they have greater experience of the double-acting engine; and they have successfully used tar and similar oils as fuel.

Although our official information with regard to the total horsepower which has actually been obtained per cylinder is a little vague, we are able by piecing together various facts which have come to our knowledge to arrive at what we think is very near the exact truth. Officially we know that one firm will guarantee 8,000 horsepower on a single shaft, but officially we do not know how many cylinders are required for that power. Officially we equally well know that that firm has carried out experiments on a 2,000 horsepower double-acting cylinder and that those experiments were successful. We may therefore fairly presume that the 8,000 horsepower would be obtained from four double-acting cylinders. We know, too, unofficially that one or two firms have already built and run an engine consisting of a number of cylinders, each of which gives 2,000 horsepower, and that such engines are intended for the German Navy. Then, too, rumors are afloat, not as far as we can find wholly without foundation, to the effect that the German Admiralty has already ordered a cruiser of about 5,000 tons which is to be fitted with motors of sufficient power to give her a speed of 25 knots. Judging, then, from the

ships in our own Navy she will have to be fitted with engines of a total of about 24,000 shaft horsepower. Putting two and two together, again we think we are perfectly justified in saying that the German Navy has now on order a cruiser which will have twin-screw Diesel engines consisting of two sets of six-cylinder double-acting engines, each set giving 12,000 horsepower, that is to say, 2,000 horsepower per cylinder. Equally, we think, we are justified in saying that these engines are not merely a design which will be evolved by the addition of five more cylinders to an already existing experimental one-cylinder engine. There is no doubt that 6,000 horsepower has already been obtained on three cranks, but we cannot say with equal certainty that a 12,000 horsepower six-crank engine has already been built. By the side of this the Vickers' 6,000 horsepower which is to be obtained in due time from six cranks looks very small; twelve cylinders in a row do not appeal to us as a matter of practical marine engineering. We feel that the present moment is a particularly appropriate one for laying special stress upon the facts which have come to our knowledge and the deductions we confidently make therefrom, in view of the Royal Commission on Fuel and Engines which is now sitting. Though doubtless the members of that Commission are perfectly well aware of the information given above, it is just as well that the man in the street should know it too, so that the actual facts with regard to the state of progress in this country for naval purposes may be in no doubt. We write thus with no idea of depreciating what has already been done in this country, but rather with the feeling of the urgent necessity that we, here, should keep pace with what is being done by our possible rivals.—"The Engineer."

## GEARED STEAM TURBINES.

The progress which has been made in the application of geared steam turbines to marine propulsion is shown by some interesting figures published in the annual report of the Parsons Marine Steam Turbine Company. No less than fifteen vessels of various types, with an aggregate of over 100,000 horsepower, are now built or building with geared turbines on the Company's system. Amongst the fifteen vessels are the two cross-channel steamers *Normannia* and *Hantonia*, built for the London and South Western Railway Co., by the Fairfield Shipbuilding and Engineering Co., three passenger steamers building for the South Indian Railway Company's new Indo-Ceylon service, an ocean-going passenger steamship, a cargo ship recently ordered by the Cairn Line of Steamships, Ltd., from Messrs. William Doxford & Sons, and several destroyers. Of the latter, one, H.M.S. *Badger*, has given very satisfactory results on the official trials, while another, H.M.S. *Beaver*, is nearing completion at Dumbarton. The *Normannia* and *Hantonia* have been on service since the spring of this year. Their machinery continues to give every satisfaction, and the economy of fuel has been remarkable. Experience with the *Vespasian* has been of the most encouraging nature, and the results obtained in that vessel have had considerable influence in the later developments.—"The Shipbuilder."

## DEVELOPMENTS IN BATTLESHIP DESIGN.

The fact that there is no possibility of arresting the development in size of battleships is once more established by two British ships to be floated this month—the *Iron Duke*, to be launched on Saturday at Portsmouth Dockyard, and the *Marlborough*, to follow from Devonport Dockyard

on the 24th inst. Both vessels mark a large step forward in every respect, and still greater is the advance decided upon in connection with the vessels which will be laid down in the slips vacated by these two. The development in size has been continuous, and has been rendered necessary in order that our ships shall excel, in each unit which constitutes fighting power, the vessels designed for other navies. It can be said with absolute confidence now, as for many years, that these ships will be superior to anything yet projected in other countries. Some statements have been made regarding the coming ships; but as it is not considered desirable in the interests of the Service that any accurate details should be published, we refrain from referring further to them than to say that such particulars as have been already published are incorrect.

Of the *Iron Duke* class there are four ships, the two in addition to those already named being the *Delhi*, under construction at Vickers' works at Barrow-in-Furness, and the *Benbow*, in course of construction at Beardmore's works at Dalmuir. All four vessels have a length of 580 feet, and a beam of 90 feet, the displacement being 25,000 tons. Since the advent of the *Dreadnought*, in 1905—seven years ago—the length has been increased by 90 feet, and the beam by 8 feet, while the displacement tonnage has been augmented by 7,000 tons. This growth has been gradual. The *Dreadnought*, of 490 feet, and 17,900 tons displacement, was succeeded by three ships of the *Bellerophon* class, of the same length and 18,600 tons. In the following year, 1907–1908, three ships of the *St. Vincent* class were laid down, and their length was 500 feet and the tonnage 19,250. Then came the *Neptune*, with the *Colossus* and *Hercules*, 510 feet in length and of practically 20,000 tons. In these ships a change was made in the arrangement of the guns. All of the ships so far had been fitted with ten 12-inch weapons, although developments had been made in the length, and therefore in the power. The five barbettes in each were located as in the *Dreadnought*—one in the center line forward, one in each fore quarter on the side, and two in the center line aft; but in the three ships of the *Neptune* class the two pairs of after guns were placed at different levels, so that all four could fire astern.

The *Orion*, which was laid down at Portsmouth in the autumn of 1909, marked a new departure in many respects. It was in this ship that the 13.5-inch gun was first fitted, and at the same time the disposition of the guns was altered. The barbettes were all arranged in the center line—two forward, one amidships, and two aft. As in the *Neptune*, those aft were arranged at different levels, and this system was also adopted for the four guns forward, so that the guns in No. 2 barbette fired forward over those in No. 1, and those in No. 4 fired aft over those in No. 5. Thus four guns were available for ahead fire and four for astern fire, while all ten could be fired on either broadside, as against eight in the preceding ships. This modification involved a considerable increase in the length of the ship, which became 545 feet, while the beam was 88 feet 6 inches, and the displacement 22,500 tons. Of this class three other ships were ordered—the *Conqueror*, *Monarch* and *Thunderer*. In the program of 1910–11 there were again four battleships—the *King George V*, the *Centurion*, the *Ajax* and the *Audacious*. In their case the arrangement of the 13.5-inch guns was as in the *Orion* class, but the length of the ship was increased to 550 feet, the beam to 89 feet, and the displacement to 23,000 tons. The program of 1911–12 provided for four battleships—of the *Iron Duke* class—and, as already mentioned, these have a length of 580 feet, a beam of 90 feet, and a displacement of 25,000 tons. The primary armament is the same as in the *King George V*—namely, ten 13.5-inch guns, and the emplacement of the guns in the ship again corresponds generally with that evolved for the *Orion*.

There has been a development, too, in connection with what have hitherto been termed torpedo-repelling guns. In the *Dreadnought* it was

considered sufficient, in view of the then range of the torpedo, to fit only 3-inch guns, but in subsequent ships the 4-inch gun was adopted, owing to the improvements in the torpedo and an increase in its range. The number of these guns fitted has varied, and changes in position have been made in order to protect them more effectually. For the most part they were accommodated in the superstructural work of the ship, behind shields. In the *King George V* class 6-inch guns were adopted, and the change now made in the case of the *Iron Duke* is the fitting of the 6-inch guns behind the broadside armor on each side of the ship. This, in some measure, is a reversion to former practice, where secondary armament was adopted. It remains to be seen, however, whether there will be any departure from the tactical standpoint that in fleet actions big guns alone should be used; that will probably depend on the range and other circumstances of the moment.

As regards armor protection, there has been steady advance, principally in the thickness of the broadside armor, and no doubt this advance will continue in the case of the new ships. At the same time the height of broadside protected by armor has been increased since the time of the *Dreadnought*, all vessels since the *Orion* having the armor carried up to the upper deck instead of to the main deck, as in the earlier ships.

As to speed, this has remained constant for battleships at 21 knots, under increasingly severe trial conditions, which give assurance that it will be easily maintained, if not exceeded, in service. Experience has not shown the need for any material difference in the arrangement of the turbines in the ships so far ordered. On each of the four shafts there is an ahead and an astern turbine. The high-pressure turbines for ahead and astern working are independent; the low-pressure ahead and astern turbines on two of the shafts are incorporated within the same casing, so that six turbine casings suffice. The revolutions throughout have been maintained at about 300 to 320 per minute, which have been found to give good results, in respect alike of weight of machinery and of propulsive efficiency. The power, however, has had to be increased in order that the speed may remain constant, notwithstanding increased displacement tonnage. Thus, in the *Dreadnought* the designed power was 23,000 horsepower; this has steadily increased to 29,000 horsepower in the case of the ships to be launched this month. Thus while the tonnage has increased by 40 per cent., the designed horsepower has only increased by 20 per cent. In other words, the ratio of power to tonnage in the *Dreadnought* was about 1.28:1, and in the ships to be launched this month 1.16:1. Experience has enabled weight to be reduced without diminishing the adequacy of steaming capacity, which is so essential with turbine-driven ships, because of the ability of the engines to take an overload under emergency.

In the *Iron Duke* there will be one mast and two funnels; the tripod system of mast will not be adopted, the mast will be stayed with wire rigging, as in the case of the *King George V*. Tanks, to counteract rolling, and the submarine telephone system will be fitted. As in the *King George V*, the officers will be berthed aft and the men forward.—"Engineering."

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## SHIPS WITH CORRUGATED HULL PLATING.

The Ericsson design of ship, having the shell-plating arranged with two longitudinal corrugations along the side between the water line and the turn of the bilge, is steadily growing in favor. The first ship—the *Monitoria*—built, like her successors, by Messrs. Osbourne, Graham & Co., Sunderland, has been running for three years; the second—the *Hyltonia*—for one year; the third was launched in May last; the fourth in July; and two others are in course of construction; while a seventh is being

built in Norway. The results of the working of the two completed ships, and the confidence disclosed in ordering the others, justifies some examination of the claims set up for the design, and it was not, therefore, inappropriate that these should be dealt with in a paper read at the Royal United Service Institution by Captain G. S. MacIlwaine, R.N. (retired). He had made several voyages in one of the ships with a view to forming an independent judgment of the merits of the case, and the aim of his paper was to secure that the system should be considered by the Admiralty. The points he put forward were: increased strength, specially called for in the case of destroyers; greater resistance to penetration by projectiles, obtainable even in the case of armored ships, especially when the ship was rolling in a heavy sea during action; and higher propeller efficiency, consequent on superior stream lines due to the corrugations. The Admiralty are no doubt fully cognizant of the features and potentialities of the design; but Captain MacIlwaine's statement of the case will be useful to Navy officials as well as to shipowners.

The *Hyltonia* is representative of the system. She is 279 feet long between perpendiculars, 39 feet 10 inches beam, 18 feet draught, with 3 feet freeboard, and carrying 3,340 tons dead weight she displaces 4,614 tons. The net registered tonnage is 1,149 tons. There are, in her case, two corrugations fore and aft, the width from the top of the upper corrugation to the bottom of the lower being 13 feet, 3 inches, and from the inner edge of the frames the corrugations project 22 inches. Forward and aft the corrugations taper to merge into the normal lines of the hull. Before deciding to build the ship the owners carried out some tests with models in a tank on Caw's pendulum system. Models of plain and corrugated ships, without other variants, were floated and attached pivotally to a pendulum first held at a known angle to the vertical by the model being secured in the tank to a cord. By the burning of the cord the model was gently released, and the amplitude of swing of the pendulum determined the comparative resistance of models. The results suggested possibilities of "a reduction in effective horsepower of from 14 to 23 per cent." The claim made is that "the space between the corrugations seems to act as a conduit pipe supplying the screw, which, in its turn, seems almost to play the part of a pump drawing a solid body of water along the ship's side in which to work." In other words, that the stream lines are better, the water being less disturbed. Captain MacIlwaine said that the blades of the propeller could be seen rotating at all speeds as clearly as through crystal. The data of actual performances showed that when the corrugations were fully immersed the slip of the propeller averaged less than 2 per cent., as against 13 per cent. for ordinarily plated ships of the same design. The average revolutions in the corrugated ship were 58, the power 630 indicated horsepower, the normal speed about 9 knots, and the fuel consumption, when loaded, 10.8 tons of Lancashire coal per day. For the same duty with the plain plated ship the power necessary was 700 to 750 indicated horsepower, and the consumption 12 tons per day.

It was claimed, too, that the corrugations add to strength. In the *Hyltonia* every alternate frame was left out under the highest classification of the British Corporation, so that the frames were 48 inches apart, as compared with 23½ inches in ordinary ships of the same size; this, and the absence of stringers, effected a saving of 50 tons in weight. Again, as tonnage measurement is from the inside of the framing, there is increased cargo space without tonnage dues being exacted, for the space given by bulging out in the corrugated plates is not reckoned. The same considerations facilitate inspection and the handling of cargo. Increased steadiness at sea and greater stability were also claimed, and results were given of sea behavior. In one voyage the deck portion of a cargo of timber was 43 per cent. of the whole, the average height of the deck cargo

being 18 feet; on another occasion the figures were 45 per cent. and 19 feet respectively, 428 tons of water ballast being carried in the tanks. It was also contended that vibration was reduced and steering made easier, while the constructional cost was about the same.—"Engineering."

## THE DUTIES OF MEMBERS OF TECHNICAL SOCIETIES.

The difficulty which the Council of the American Society of Naval Engineers experiences in procuring for publication more than a very small percentage of the vast fund of new and important technical information known to be available among the large number of members and associates of the Society, causes it to reprint the following from "The Engineer," with a hope that it may recall to notice the purposes of the Society as set forth in Section 2 of its By-Laws, and the duties of members and associates, as expressed therein.

The Society can not be expected to prosper duly, nor its Journal to possess a tithe of its potential value to the naval engineering profession, as long as such a large majority of its membership persists in the passive attitude referred to in the article.

Mr. E. Hall-Brown, president of the Institution of Engineers and Shipbuilders in Scotland, in his opening address, took occasion to administer what may be considered a mild admonition to the members upon a growing disinclination which seems to be manifesting itself in the matter of full and free discussion of papers submitted, and of neglecting the duty of making contributions to the Institute dealing with current questions of general interest. The *raison d'être* of such corporations is to facilitate free interchange of experience and opinion to the benefit of the Institution as a whole, and its members individually, by the pooling of their respective experiences for the general good. If a narrow, secretive attitude be assumed with regard to valuable information, then it follows that such Institutions must in time become defunct, in reality if not in mere membership. A professional institution in a moribund condition is not inspiriting to contemplate, for it indicates a spirit among its members which is antipathetic to progress, and a serious bar to the healthy spread of professional knowledge. The tendency remarked upon is not peculiar to the members of the particular institution to which the speech was addressed, and it is in marked contrast to the condition of things in some societies of more recent growth, as instanced by the contributions of Captain Dyson and D. W. Taylor to their technical bodies in America, where very valuable information has been freely and fully put at the disposal of the general body of engineers. It is quite understandable that neither an individual nor a firm which has spent large sums of money upon investigation work for the elucidation of some obscure feature of its profession should be anxious at once to present the results openly to others who are in business competition with it, and no one would blame it for not doing so, but there are many experiences which are not so obtained, are of great importance, and would be of general value. It is easier to obtain such information about undertakings which have been conspicuously successful than about others which have not fulfilled so completely the expectations of their authors, and yet if professional men took a large enough view to enable them to give these as well they would be found to be often more valuable than those which follow smoothly an expected direction of development.

What are the reasons which might cause such a disinclination to give and take in professional matters as it has been suggested now exists? Is

it that there is a decadence in the spirit which evoked the foundation and has sustained the being of professional institutions? We should not like to think so. There can be no doubt that sharp competition, whether between Governments with regard to their naval and military supremacy, or between business men who have to fight for orders in the keenest markets, has the effect of making each of them desirous of retaining any advantage of knowledge or experience that he may possess, but this does not sufficiently explain the position. We are more inclined to the view that the rapid and fundamental changes which have been made, and are still taking place in engineering, shipbuilding, and metallurgy, are responsible for much of the backwardness which prevents the best men expressing opinions upon or reporting results of ventures which, large and important as they may be, are often more or less tentative and experimental. Take the case of the profession to which Mr. Hall-Brown addressed himself—that of shipbuilding and engineering. What enormous changes have taken place in the short time which has elapsed since a compound or triple-expansion steam engine of reciprocating type, using steam from ordinary Scotch or haystack boilers, was the standard of marine machinery, while a speed of 14 or 15 knots represented practically the maximum speed of ordinary vessels? Now we have steam turbines driving direct or through gearing and of more than one type, oil engines, electric propulsion, water-tube boilers fired with coal or oil, or both, high-pressure steam, superheaters, refrigerators, hydraulic engines, steam-steering engines, etc., to say nothing of the greatest bogie of all, the screw propeller, which has to be made to suit the vagaries of speed of all these variations of prime mover. It is only to be expected that the vast amount of new ground covered involves a corresponding amount of exploratory work, much of which is expensive and difficult of accomplishment; at the same time it involves its authors in a great deal of unavoidable labor, which reduces the amount of time and energy at their disposal for the more general advancement of their profession; it may also make them chary of publicly expressing opinions and relating experiences which will be put in a more or less permanent form, but which they feel may be rendered nugatory tomorrow or the day after because of the state of transition in which they find themselves. The disinclination is therefore easy to understand. It would not be pleasant to think that lack of interest or narrowness of outlook was the cause to which should be attributed reduced enthusiasm for professional conference or intercourse; it is probably rather the result of somewhat exceptional circumstances; and yet is it not this very fact that should make contribution to and enlightenment from such societies at once a duty and a benefit?

It is true that in the beginnings of any new development only the few usually have anything authoritative to impart; many of their *confrères* are able and willing to lend their mental aid and professional skill to the solution of knotty problems, but if they do not possess the information upon which to work the value of their intellectual concentration is lost. To those who see farthest it is evident that the wider view is the saner view, for no man can exactly estimate the mental value of his brother, nor can he guess from whence the next idea is to spring in the process of evolution; but it demands from the possessors of knowledge the wisdom and generosity to give without hope of getting back personally, though it often happens that bread thus cast on the waters comes back after many days, for he who gives on one particular subject gets from others all the varied contributions and discussion upon allied subjects, as well as probable help in the solution of his own difficulties. We refuse to believe that there is less generosity in the minds of engineers today than there has been in the past, or that the commercial spirit can utterly quench loyalty to the traditions of their fathers, who gave and received and made their professions what they are by means of the mental attrition

of their fellows, which modifies crude ideas and rounds up immature thoughts as effectively in the technical sphere as it softens acerbities and rounds up character in the purely social side of life.

## THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT.

### MAJORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION.*

During the past few years a number of striking phenomena, in connection with industrial management, must have become evident even to the most superficial observer. The more important are:

(*a*) The widespread, popular interest in the subject which had its rise in a statement made before the Interstate Commerce Commission, in a hearing on the matter of proposed advances in freight rates by carriers. An attorney for the shippers stated on November 21, 1910, that it was estimated that by the application of newly discovered principles of management "in the railroad operation of this country an economy of $1,000,000 a day is possible," and further that these principles can be applied with equal success "in every form of business activity." This popular interest is shown by the great number of articles published in the daily papers and popular magazines, mediums that give but scant attention to technical subjects, except of the most striking nature.

(*b*) The suddenly intensified interest in the subject on the part of employers and business executives in many lines of activity, shown by lectures, addresses, professional papers and reports presented to their associations.

(*c*) The opposition of labor unions to the newer methods of management, shown by statements of labor leaders, in a few instances by strikes, and by an attempt to prohibit by law the use of some of these methods in Government shops.

(*d*) Governmental recognition of the matter shown by the appointment of a special committee of the House of Representatives to investigate systems of management in Government arsenals and shops, which reported in March, 1912; by the appointment of a civilian board by the Secretary of the Navy to investigate management in the navy yards which reported in July, 1911; and by Senate bill S. 6,172, now in committee, which is intended to prohibit time study and the payment of premiums or bonus on Government work.

(*e*) The rapidity with which literature on the subject has accumulated. One directory of books on business management lists 500 titles, and states that 75 per cent. of them have been written within five years.

(*f*) The formation of two societies having as an aim the furtherance of the application of the principles of management.

(*g*) The separation of persons interested in the matter into two camps, one of enthusiastic advocates, the other of vigorous opponents of what is called the new element in management.

(*h*) The unquestionable proof of the advance that can be made in unskilled work, as shoveling material, and in ancient trades, as bricklaying, by the application of the principles of management. This is the most striking phenomenon of all.

### THE PRINCIPLES OF MANUFACTURE.

2. Before defining the element in the art of management that has given rise to these phenomena, it is necessary to review briefly the beginnings of modern industry. This gives a historical setting from which the present can be more truly judged.

* The American Society of Mechanical Engineers. Subject to revision.

3. Modern industry is stated by some writers to have begun in 1738 when John Wyatt brought out a spinning machine. Others place the period as between 1750 and 1800, when the power loom and steam engine came into being. It was marked by the development of labor-saving machinery. It was brought about by the change from handicraft to manufacture.

4. Early British economists held that the application of the principle of *division of labor* was the basis of manufacture. From Adam Smith's "Wealth of Nations," 1776, we quote:

"This great increase of the quantity of work which, in consequence of the division of labor, the same number of people are capable of performing, is owing to three different circumstances; first, to the increase of dexterity in every particular workman; secondly, to the saving of the time which is commonly lost in passing from one species of work to another; and lastly, to the invention of a great number of machines which facilitate and abridge labor, and enable one man to do the work of many."

5. Charles Babbage, the great British mathematician and mechanician, believed that from the above-quoted statement the most important principle was omitted. This omission he supplies as follows in his "Economy of Machinery and Manufacture," 1832:

"That the master manufacturer, by dividing the work to be executed into different processes, each requiring different degrees of skill and force, can purchase exactly that precise quantity of both which is necessary for each process; whereas, if the whole work were executed by one workman, that person must possess sufficient skill to perform the most difficult, and sufficient strength to execute the most laborious, of the operations into which the art is divided."

6. It appears, however, that another principle is the basic one in the rise of industry. It is the *transference of skill.* The transference of skill from the inventor or designer to the power-driven mechanism brought about the industrial revolution from handicraft to manufacture. It will be necessary to refer to this principle frequently throughout this report, in showing the meaning and position of management in industry.

7. No better single illustration of the application of this principle can be found than in the invention of the lathe slide rest by Henry Maudsley in 1794. This has been ranked as second only to the steam engine in its influence on machinery building, and thus on industrial development. The simple, easily-controlled mechanical movements of the slide rest were substituted for the skillful human control of hand tools. So complete has been this transference of skill that today hand tooling is a vanished art in American machine shops. Very few lathe hands can chase a thread with hand tools, yet all can cut good threads on an engine lathe, thanks to the slide rest. After the traditional skill of a trade, or the special, peculiar skill of a designer or inventor, has been transferred to a machine, an operator with little or no previously acquired skill can learn to handle it and turn off the product.

8. An example of the extent to which this transference of skill is carried today is presented by the shoemaking industry. The United Shoe Machinery Company builds some 400 machines used in shoe manufacture. These are so highly organized that the greater part of shoe-shop operatives are unskilled except in a single readily-mastered detail of the work. The skill in shoemaking is now in the mechanical equipment of the shops. This transference is a development of the past 50 years.

9. James Nasmyth, a British engineer, inventor of the steam hammer, has this to say in 1851 of the application of this principle in his own works: "The characteristic feature of our modern mechanical improvements, is the introduction of self-acting tool machinery. What every mechanical workman has now to do, and what every boy can do, is not to work himself, but to superintend the beautiful labor of the machine. The

whole class of workmen that depend exclusively on their skill is now done away with."

10. Methods of analyzing and recording operations were early developed. Adam Smith records the divisions of the work of manufacturing pins, listing 11 operations. Charles Babbage gives a table (see Appendix No. 1, Table 1) from a French investigator, showing the number of operations, time for each, cost of each, and expense of tools and material for making pins in France in 1760. He gives a similar table for English manufacture in his day (see Appendix No. 1, Table 2, from Economy of Machinery and Manufacture, 1832).

11. He further comments on the use of the watch to time operations. We quote from his instructions to one making such observations and using a skeleton form that he recommends: "In filling up the answers which require numbers, some care should be taken; for instance, if the observer stands with his watch in his hand before a person heading a pin, the workman will almost certainly increase his speed, and the estimate will be too large. * * * The number of operations performed in a given time may be frequently ascertained when the workman is quite unconscious that any person is observing him. Thus the sound made by the motion of a loom may enable the observer to count the number of strokes per minute, even though he is outside the building in which it is contained."

12. M. Coulomb, the noted French physicist (1736–1806), who had great experience in making such observations, cautions those who may repeat his experiments against being deceived by such circumstances. We translate a single quotation: "I pray (says he) those who wish to repeat them (the experiments) if they have not time to measure the results after several days of work, to observe the workmen at various times during the day without their knowing that they are being watched. We cannot be too well warned of the danger of self-deception in computing either the speed or the effective time of work through an observation of a few minutes."

13. Thus we see the application of the principle of *transference of skill* at the basis of the development of the industry, and an early appreciation of the value of the detailed study of operations in making that transference more complete. But the machine was the viewpoint. It was looked upon as the producing unit. Combined and contrasted with this was a lack of knowledge of scientific principles and their sure application. Charles Babbage treats of this forcefully. We quote:

"There is perhaps no trade or profession existing in which there is so much quackery, so much ignorance of the scientific principles, and of the history of their own art, with respect to its resources and extent, as is to be met with amongst mechanical projectors."

14. In the same vein he emphasizes the need of accurate drawings as if having in mind the poor quality of the work from the average draftsman of his day: "It can never be too strongly impressed upon the minds of those who are devising new machines (says he) that to make the most perfect drawings of every part tends essentially both to success of the trial, and to economy in arriving at the result."

15. He further points out that there is another important factor in successful industry, in addition to machinery. We read that "in order to succeed in a manufacture, it is necessary not merely to possess good machinery, but that the domestic economy of the factory should be most carefully regulated."

16. These quotations foreshadow modern methods of thinking out the work in advance and transfering this thought to the workmen. The subsequent development has had the effect of advancing still further the division of labor, and beginning the division of thought. The drafting room presents the first example of the trend, in its collection of engi-

neering data, in its prediction of results and the formation of staff organization.

17. But from the period of the last quotation almost to the present there has been no change in the basic principles discovered and applied in industry. There has been nothing but an extension of those already known. The place of greatest advance has been in the drawing room. The art of machine design has been greatly developed. The last half of the last century saw a tremendous increase in inventions, a tremendous furtherance of the application of transference of skill to machines and tools. The skeleton of an industrial organization of this period, one that was too large for a single executive to manage, consisted of a designing department and a production department, each with a head responsible to the manager.

18. The first of these, the one that was the means of embodying skill in the machinery and tools of production, was highly developed and organized. Experiment, research and detailed study were constantly resorted to, to aid in reaching the desired result. The work was highly specialized and the employees highly paid. Not infrequently the manager or chief executive devoted much of his own time to this part of the business.

19. The production department presented a contrasting condition. The workmen were given the tools and machines designed in the drawing room and using their own unaided skill were expected to produce work of the desired quality and quantity. Except in rare instances no effort was made to transfer the skill of the management to the production department and the employees, or to undertake the division of executive thought. Very little consideration was given to the workmen as a producing unit.

FEATURES OF THE CHANGE.

20. Within the past 20 or 25 years certain changes have taken place in the attitude of many production managers toward the problems that they face and the forces and means that they control. An increasing amount of attention is being given to the worker. An early evidence was the development of profit-sharing, premium and bonus systems to reward increased effort and output. There followed welfare work, industrial betterment movements, the adoption of safeguards and regulations to minimize industrial accidents, the substitution of the principle of accident compensation for employers' liability and an improvement in the physical surroundings and conditions of factories. All of these tendencies have been fostered and to a great extent initiated by employers. But even today these are by no means generally adopted.

21. Another tendency, less pronounced in character, has as its object the improvement of the personal relations between employee and employee and between employee and employer. It is an effort to establish the best of factory working conditions in those things not physical in nature, to develop and maintain a shop atmosphere free from all harassing and hindering influences. It is an attempt to make use of the results of experimental psychology, in improving working conditions.

22. But the most important change and one that comprehends the others, is in the mental attitude toward the problems of production. The tendency is toward an attitude of questioning, of research, of careful investigation of everything affecting the problems in hand, of seeking for exact knowledge and then shaping action on the discovered facts. It has developed the use of time study and motion study as instruments for investigation, the planning department as an agency to put into practice the conclusions drawn from the results of research, and methods of wage payment which stimulate coöperation.

23. All of these changes have affected the production department much

more than the designing department. The effect is to extend the principle of transference of skill to production, so that it completely embraces every activity in manufacture. The skill of the management is consciously transferred to all of the operations of the factory. This extension is expressed by these phrases; the drawing room is the planning department of design, and the planning department is the drawing room of production.

NATURE OF THE COMMITTEE'S INVESTIGATION.

24. To obtain information on present conditions your committee wrote to the recognized experts, to executives of plants in many lines of industry, to students of industrial problems, and has had many interviews with men in these various fields. The response to our requests has been in the main most generous. We are deeply indebted to the information thus received for a large portion of the following sections of this report. We are glad to take advantage of this opporunity to express our gratitude to all those who have given aid.

25. Throughout the following pages there is a plentiful use of illustrative quotations. Many of these are taken from correspondence resulting from our investigations. Others are from the mass of literature mentioned in Paragraph 1*e*.

26. On some points diametrically opposed views have been expressed. In such cases we have presented both. In no case has credit been given for these views or quotations, as the information was solicited in confidence.

DEFINITION OF THE NEW ELEMENT IN THE ART OF MANAGEMENT.

27. Requests for a definition of the new element in the art of management brought forth a difference of opinion as to its existence. The opposed view is given in the following quotations:

"I am not aware that a *new* element in the art of management has been discovered———."

"There have been no new discoveries in scientific management of industrial institutions. Common-sense men have used common-sense methods always. The term 'scientific management' is a catch-word which assumes that industrial institutions have not been scientifically managed—which is not the case. My experience and the experience of my friends has been that there has been no new element injected into the art of management."

"In the writer's opinion there is very little that is new about it (the art of management). There is hardly any part of it that has not been practised by managers for the past 100 years. The trouble is there are not enough managers with sufficient initiative to set the system moving properly."

"——— the problem presented is not the adoption of something entirely new; but rather the extension to every detail of our work of something which we have already tried."

28. Turning now to the other side of the question, from a large number of definitions of this new element we select the following as very nearly conveying, taken together, the complete conception as our investigation has disclosed it:

"The best designation of the new element I believe to be 'scientific management.' This term already has been adopted quite generally and, although frequently misused, carries with it the fundamental idea that the management of labor is a process requiring thorough analytical treatment and involving scientific as opposed to 'rule of thumb' methods."

"The writer ventures to define the new element briefly, but broadly, as:

The critical observation, accurate description, analysis and classification of all industrial and business phenomena of a recurring nature, including all forms of co-operative human effort and the systematic application of the resulting records to secure the most economical and efficient production and regulation of future phenomena."

"Stripped of technicalities the method of the modern efficiency engineer is simply this: First, to analyze and study each piece of work before it is performed; second, to decide how it can be done with a minimum of wasted motion and energy; third, to instruct the workman so that he may do the work in the manner selected as most efficient."

"The Taylor System is not a method of pay, a specific ruling of account books, not the use of high-speed steel. It is simply an honest, intelligent effort to arrive at the absolute control in every department, to let tabulated and unimpeachable fact take the place of individual opinion; to develop 'team play' to its highest possibility."

"As we conceive it, scientific management consists in the conscious application of the laws inherent in the practise of successful managers and the laws of science in general. It has been called management engineering, which seems more fully to cover its general scope than a science."

29. These quotations convey the ideas of a conscious effort to ascertain and study facts and systematically to apply them in instructing the workmen and in controlling every department of industry. Setting these against the underlying principle of the transference of skill we conceive the prominent element in present-day industrial management to be: *The mental attitude that consciously applies the transference of skill to all the activities of industry.*

30. Here emphasis is placed upon the word *all* for, as shown in Pars. 17 and 18, the restricted application of this principle to machines and tools has been highly developed for a long period. But its conscious application in a broad way to the production departments, and particularly to the workmen, we believe has been made during the last quarter century.

## RISE OF THIS MENTAL ATTITUDE.

31. The rise of this change of attitude in regard to industrial management is shown in the papers on the subject in the Transactions of this Society. These are 16 in number and are listed in Appendix No. 2 of this report. The period covered is from 1886 to 1908. The practice upon which several were based extended over a number of years before the paper was presented. Papers on accounting have been excluded.

32. The first, No. 207, classifies *management of works* as a modern art having a vast amount of accumulated experience, points out that the executives must have "a practical knowledge of how to observe, record, analyze and compare essential facts in relation to * * * all * * * that enters into or affects the economy of production and the cost of the product," and makes a plea for the interchange of management data.

33. Eight following papers, Nos. 256, 341, 449, 596, 647, 928, 965 and 1012, deal with methods of wage payment, showing the increasing attention given to the workmen during this period. Of these methods the "premium plan" described in paper No. 449 has an extensive use today in machine shops. It probably ranks third, in extent of use, being exceeded by day work and piece work in the order named. Paper No. 647 outlines elementary rate fixing; that is, the minute study of each detail of each operation. From this, motion study and time study have grown. The "bonus system" of paper No. 928 also has an extensive use, probably ranking fourth.

34. Paper No. 1003, "Shop Management," is the first complete presentation of the subject. This paper with the subsequent writings of its author, stands today as the only comprehensive outline of industrial man-

agement. Papers Nos. 1001, 1002, 1010, 1011 and 1115, are amplifications of certain features of No. 1003 and are based on the same practice.

35. Paper No. 1221 deals with the training of workmen, and outlines practical, tested methods of bringing about the all-important transference of skill.

LABOR-SAVING MANAGEMENT.

36. Since these papers were presented, and during the development of popular interest in the subject, the term "scientific management" has been generally and loosely applied to the new system and methods. This is commonly taken to mean that there is a science rather than an art of management. A truer interpretation is that it means management using scientific methods, these being taken largely from the sciences of physics and psychology.

37. The expression "labor-saving management" better conveys the meaning of the movement. It has the further advantage of being easily and surely understood because of its strict analogy with the term "labor-saving machinery." It is no chance that puts these two terms labor-saving machinery and labor-saving management, in conjunction, for the first is the past development and the second the present trend of industry, and they will be closely and inevitably associated in the successful manufacturing of the future. Throughout the following pages of this report the terms "industrial management" and "labor-saving management" are used, the first to denote the subject broadly, the second the newer attitude.

THE REGULATIVE PRINCIPLES OF INDUSTRIAL MANAGEMENT.

38. The lack of accurate thinking and clear expression in regard to management are nowhere better shown than in many of the statements of the so-called principles. These can be divided into two classes, personal characteristics of managers and mechanical means of applying. It is evident that neither can show us the way in which the activities of industry are to be regulated.

39. In our investigation preparing for this report, one correspondent writes as follows:

"The regulative principles of management along scientific lines include four important elements:

"(*a*) Planning of the processes and operations in detail by a special department organized for this purpose.

"(*b*) Functional organization by which each man superintending the workman is responsible for a single line of effort. This is distinctly opposed to the older type of military organization, where every man in the management is given a combination of executive, legislative and judicial functions.

"(*c*) Training the worker so as to require him to do each job in what has been found to be the best method of operation.

"(*d*) Equable payment of the workers based on quantity and quality of output of each individual. This involves scientific analysis of each operation to determine the proper time that should be required for its accomplishment and also high payment for the worker who obtains the object sought."

40. Another correspondent finds the solution of problems of management in the observing and regulating of three classes of industrial phenomena:

"(*a*) The economic results of different arrangements and forms of materials and operations upon them, either to produce equipment or product. This covers the whole field of recorded experience from invention and design of product and tools down through the successive shop processes to ultimate finished product and its tests in service. It is the ob-

ject of the scientific method to make the best of this experience, in its essential details, readily available for all concerned, and to see that it is actually absorbed and put in practice.

"(*b*) The economic results of varying executive methods for effectively directing human efforts as a whole in the use of the above experience. This covers the entire field of building up, co-ordinating and controlling the supervising organization of a plant with its statistical and recording systems.

"(*c*) The economic results of steps taken to raise the industrial efficiency of the individual worker in every grade of service. This covers the whole problem of labor reward, intensified ability, conserved energy and the general relations of employer and employee."

41. We have pointed out that the underlying principle, that is, cause in the widest sense, the application of which has built up modern industry, is the transference of skill. This basic principle is put into effect on the management side of all industrial activities, through three regulative principles which sum up the ideas in the above quotations. Pars. 39 and 40. These have been concisely stated as*: (*a*) the systematic use of experience; (*b*) the economic control of effort; (*c*) the promotion of personal effectiveness.

42. The first includes the use, in all essential detail, of traditional knowledge, personal experience and the results of scientific study on the part of the executive force. It implies the accumulation and use of records and the setting up of standards.

43. The second includes the division and subsequent co-ordination of both executive and productive labor; the planning of single lines of effort, the setting of definite tasks and the comparison of results; and the effective training of the workers. It implies the previous acquisition of skill by the executives.

44. The third includes a definite allotment of responsibility and the adequate, stimulative encouragement and reward of both executive and productive labor; the development of contented workers, and the promotion of their physical and mental health. It implies the most thorough comprehension of the human being.

### THE PRACTICE OF MANAGEMENT.

45. As labor-saving management springs from a change in mental attitude, the beginning of its practice should be with the persons having the final responsibility, the proprietors of closely-owned business, the directors of larger establishments, or the officials having charge of Government works. Before any changes are made, such men should clearly understand the viewpoint from which all of the managerial work is to be done, the principles that are to be applied, the general method of their application and the results expected.

46. A similar mental attitude must be fostered among all the members of the executive force and a period of training for them begun. This may include a redistribution of function and responsibility, and will include a detailed study of production by scientific methods. This is the period of division of thought, training of the management staff and setting-up standards of performance. This must be carefully performed before there can be effective transference of skill to the workers in the production departments.

47. The usual conception of modern management is that it affects the workmen most of all, tending to stimulate them to turn out increased production to their possible hurt. This is wrong. If the principles outlined are followed, the executive, or non-producing labor is the most affected.

* "American Machinist," vol. 36, p. 857, The Principles of Management, by Church and Alford.

Its individuals are compelled to study, plan and direct. They must acquire knowledge and skill in order to transfer it. It is a system of management that forces the executives to manage.

48. This being so, the introduction of modern management in a plant must be made slowly. The causes of most so-called failures are principally two: a failure of the executives to acquire the vital mental attitude and too great haste in application. The latter seems to be the dominant one. Your committee feels compelled to emphasize the danger of attempting to hurry any change in methods of management. Each step of the work should be made permanent before the next is begun.

49. We have examined records of production which clearly show a lessening of individual output among workers who had been trained for some time and had achieved good results as soon as untrained workers were put with them, thus lessening their share of personal supervision. Later the original standard of production was again reached, but the results seemed to be directly proportional to the amount of skillful supervision, during a lengthy period of training.

50. After those who are to operate the new methods have acquired the necessary knowledge and established sufficient standards, the work of putting these into effect can be begun. This means the fixing of the best attainable working conditions and giving each worker definite tasks with an adequate reward to each one who attains to the standard set. This part of installing the methods must be accomplished with tact and patience, remembering that leadership and example are powerful aids in bringing about enthusiastic co-operation.

51. The training of the workers is essential in this part of the application. This must be far more than mere demonstration, the mere showing that a thing can be done. It must be patient teaching and help until the required degree of dexterity or skill is acquired, that is, up to the habit stage. It is evident that such work cannot be hurried.

52. Such, broadly, are the three steps in the practice of management. It is now necessary to investigate the internal elements of permanence in such methods. If the proper mental attitude is once taken, we believe it will never be given up. This is substantiated by a few cases when early attempts to improve management were failures and the methods abandoned. Later, however, other attempts were made with substantial success. The mental attitude outlived the failure. Thus in a given industrial organization this feature would not be lost except by a loss of the executive staff.

53. The permanence of records of performance and standards needs only to be mentioned to be appreciated. Once set up in an industry, disaster is invited if they are disregarded.

54. To these is added a third in the nature of a spur from the working forces to the managing force. An adequate reward is one of the essentials. Whatever disturbs the mechanism of production interferes with the earning of the rewards. The workers at once object, pointing out the trouble and insisting that it be rectified. The management is spurred to keep all conditions up to the fixed standard. Examples of this action have been brought to the attention of your committee.

55. The practice as outlined, while built upon fixed standards and procedure, is by no means rigid and inflexible as has been alleged. The design and construction of labor-saving machinery is carried on with a multiplicity of different details. Labor-saving management should likewise use a variety of details suited to the requirements of different industries and plants. There can be nothing fixed in such human endeavor except the underlying principle. As a simple matter of fact we have found different methods, details and nomenclature in use in different plants. Many efforts have undergone marked change and development since first installed. Further, this idea of rigidity is repudiated by some of the foremost management experts.

56. In Par. 39 is emphasized the need of a scientific study of everything connected with production. The methods used are adapted from the research laboratory. But the purpose of their use is changed. The scientific investigator uses his laboratory to discover facts. Their discovery and declaration is his end and aim. The management investigator uses laboratory methods to discover facts for immediate use. The end and aim is utility. This is the test of industry. It is therefore unwise and in fact detrimental to carry investigations to an extreme. Enough facts must be observed to shape intelligent action. Persons having time study and motion study in charge should possess that rare, intuitive, human quality that causes its possessor to know when enough observations have been collected to form a sound working conclusion.

57. The position of the expert in the practice of management is more clearly seen as experience increases. The element of mystery has already departed. This is to be welcomed for it means the downfall of mere "systematizers." One of the unfortunate features of this great movement has been the rise of alleged experts who have been ready to promise extravagant results if they were allowed to systematize an industrial plant. The test which their work cannot meet is the one of permanence.

58. An industrial manager who has had signal success in directing large enterprises sums up the more undesirable characteristics of systematizing practice as:

"(*a*) The publication and quotation of statistics regarding gains made through the use of particular systems, without a frank statement of the degree of inefficiency of the plants before reorganization.

"(*b*) The failure to view the plant from the investor's standpoint rather than as a laboratory offering opportunities for interesting and expensive experience.

"(*c*) The failure to admit that every application of past solutions to unstudied new and different conditions *is* an experiment.

"(*d*) The waste of time and money on problems that will yield to scientific treatment but which do not recur often enough to justify such a solution.

"(*e*) The undervaluing of effective leadership in management and consequent lack of permanency in results.

"(*f*) The overvalue of emasculated 'system' leading to a curious non-responsibility on the part of any person for the total result.

"(*g*) The frequent assumption that the treatment of the problems of similar plants should be identical.

"(*h*) The failure to properly appraise in a growing concern the value of its internal asset of 'good will.'

"(*i*) The imperfect analysis and appreciation of the human factor in industry, with a consequent failure to reckon patiently with 'habit' and 'inertia' and a tendency to hasty 'substitution,' bringing about the breaking up of valuable organization."

59. The real expert concentrates on the facts of a given problem, and from a wide experience in analysis, co-ordination and practical responsibility works out a solution by scientific methods, suited to the material and human factors involved. The tendency is for him to do less of the detail work of installation, but to train and direct the persons who are permanently to manage. This is a true process of transference of skill.

### STATISTICAL DATA.

60. Your committee hoped to present statistics on the extent to which labor-saving management is in use. This could not be realized. Many industrial managers whom we have addressed have not honored us with their confidence in this direction. In fact, it seems as if a secretive stage

is now with us. There are two reasons for withholding such information. The first is identical with the one that has developed "trade secrets" and secretiveness in regard to machines, tools and processes, the desire to keep things of value away from competitors. The second is a belief that in the minds of some persons a reflection is cast upon the ability of the executives of an industrial establishment if outside experts are employed. Frequently a system of management is referred to as the development of some one in the organization, although it was installed by a management expert, employed for the purpose.

61. Some idea of the variety of the industries in which labor-saving management is in use can be gained from Appendix No. 3, which lists a total of 52.

BROAD RESULTS OF LABOR-SAVING MANAGEMENT.

62. In cases where the use of labor-saving management can be considered a success, the broad results have been: a reduced cost of product; greater promptness in delivery with the ability to set and meet dates of shipment; a greater output per worker per day with increased wages; and an improvement in the contentment of the workers. This last item is shown by the fewness of strikes under the new management, and in the refusal of those working under the changed conditions to join in a strike of their fellows in the same plant who were not working under the new methods. This last-mentioned situation has arisen a number of times. In one case an attempt was made to strike a room where about one-half of the operators were under the new conditions. These refused to go out; the rest went.

63. These results indicate certain advantages to both employer and employee. But it is charged that the movement has not yet entirely justified itself from the economic viewpoint, for it has not reduced the cost of product to the consumer. The implication is that its possibilities will not be realized until employers, employees and the public are alike benefited. With this view we are in most hearty accord. Labor-saving machinery has brought the comforts that we all enjoy today. Labor-saving management promises to extend those comforts. Where properly administered it is conserving labor and is thus contributing to the good of society at large, and although the benefit to the consumer may not yet be generally felt, it has already developed to a certain extent and will continue to develop as the natural result of increased production.

J. M. DODGE, *Chairman*,
L. P. ALFORD, *Secretary*,
D. M. BATES,
H. A. EVANS,
WILFRED LEWIS,
W. L. LYALL,
W. B. TARDY,
H. R. TOWNE,
*Members Sub-Committee on Administration.*

APPENDICES.

*Appendix No. 1.*

64. The following tables are taken from a book published in 1832, Economy of Machinery and Manufacture, by Charles Babbage. Table 1 gives the cost and operations in detail in manufacturing 12,000 No. 6 pins in France in 1760. The observations were made by M. Perronet.

65. Table 2 gives similar data for manufacturing pins in England in the time of the author. The size is "eleven," of which there are 5,546 to the pound.

TABLE 1.—OPERATIONS IN PIN MANUFACTURE IN FRANCE IN 1760.

| | Name of the process. | Time of making 12,000 pins, hrs. | Cost of making 12,000 pins, pence. | Workman usually earns per day, pence. | Expense of tools and materials, pence. |
|---|---|---|---|---|---|
| 1 | Wire | ..... | ..... | ..... | 24.75 |
| 2 | Straightening and cutting. | 1.2 | 0.5 | 4.5 | ..... |
| 3 | Coarse pointing | 1.2 | 0.625 | 10.0 | ..... |
| | Turning wheel* | 1.2 | 0.875 | 7.0 | ..... |
| | Fine pointing | 0.8 | 0.5 | 9.375 | ..... |
| | Turning wheel | 1.2 | 0.5 | 4.75 | ..... |
| | Cutting off pointed ends | 0.6 | 0.375 | 7.5 | ..... |
| 4 | Turning spiral | 0.5 | 0.125 | 3.0 | ..... |
| | Cutting off heads | 0.8 | 0.375 | 5.625 | ..... |
| | Fuel to anneal heads | ..... | ..... | ..... | 0.125 |
| 5 | Heading | 12.0 | 0.333 | 4.25 | ..... |
| 6 | Tartar for cleaning | ..... | ..... | ..... | 0.5 |
| | Tartar for whitening | ..... | ..... | ..... | 0.5 |
| 7 | Papering | 4.8 | 0.5 | 2.0 | ..... |
| | Paper | ..... | ..... | ..... | 1.0 |
| | Wear of tools | ..... | ..... | ..... | 2.0 |
| | | 24.3 | 4.708 | | |

*The expense of turning the wheel appears to have arisen from the person so occupied being unemployed during half his time, whilst the pointer went to another manufactory.

TABLE 2.—OPERATIONS IN PIN MANUFACTURE IN ENGLAND ABOUT 1830.

| | Name of the process. | Workmen.* | Time of making 1 pound of pins, hours. | Cost of making 1 pound of pins, pence. | Workman earns per day. s. | d. | Price of making each part of a single pin, in millionths of a penny. |
|---|---|---|---|---|---|---|---|
| 1 | Drawing wires | Man | 0.3636 | 1.2500 | 3 | 3 | 225 |
| 2 | Straightening the wire | Girl | 0.3000 | 0.1420 | 0 | 6 | 26 |
| | | Woman. | 0.3000 | 0.2840 | 1 | 0 | 51 |
| 3 | Pointing | Man | 0.3000 | 1.7750 | 5 | 3 | 319 |
| 4 | Twisting and cutting the heads | Boy | 0.0400 | 0.0147 | 0 | 4½ | 3 |
| | | Man | 0.0400 | 0.2103 | 5 | 4½ | 38 |
| 5 | Heading | Woman. | 4.0000 | 5.0000 | 1 | 3 | 901 |
| 6 | Tinning, or whitening | Man | 0.1071 | 0.6666 | 6 | 0 | 121 |
| | | Woman. | 0.1071 | 0.3333 | 3 | 0 | 60 |
| 7 | Papering | Woman. | 2.1314 | 3.1973 | 1 | 6 | 576 |
| | | | 7.6892 | 12.8732 | | | 2,320 |

*Number of persons employed: Men, 4; women, 4; children, 2. Total, 10.

*Appendix No.* 2.

66. Following is a complete list of papers published in the Transactions of The American Society of Mechanical Engineers dealing with industrial management:

| No. | | | |
|---|---|---|---|
| 207 | The Engineer as an Economist | Henry R. Towne | 1886 |
| 256 | A Problem in Profit Sharing | Wm. Kent | 1887 |
| 341 | Gain Sharing | Henry R. Towne | 1889 |
| 449 | The Premium Plan of Paying for Labor | F. A. Halsey | 1891 |
| 596 | The Relation of the Drawing Office to the Shop in Manufacturing | A. W. Robinson | 1894 |
| 647 | A Piece-Rate System | Fred. W. Taylor | 1895 |
| 928 | A Bonus System for Rewarding Labor | H. L. Gantt | 1902 |
| 965 | Gift Propositions for Paying Workmen | Frank Richards | 1903 |
| 1001 | The Machine-Shop Problem | Charles Day | 1903 |
| 1002 | A Graphical Daily Balance in Manufacture | H. L. Gantt | 1903 |
| 1003 | Shop Management | Fred W. Taylor | 1903 |
| 1010 | Slide Rules for the Machine Shop as a Part of the Taylor System of Management | Carl G. Barth | 1904 |
| 1011 | Modifying Systems of Management | H. L. Gantt | 1904 |
| 1012 | Is Anything the Matter with Piece Work? | Frank Richards | 1904 |
| 1115 | A History of the Introduction of a System of Shop Management | James M. Dodge | 1906 |
| 1221 | Training Workmen in Habits of Industry and Co-operation | H. L. Gantt | 1908 |

*Appendix No.* 3.

67. Following is a list of the industries in which some form of labor-saving management has been installed:

Book binding
Building construction
Carriage and wagon building
Construction and repair of vessels (navy yards)
Fire-arms and ordnance
  Rifles
  Gun carriages
Machinery building
  Automobiles
  Agricultural implements
  Coal-handling machinery
  Electrical machinery
  Founding, iron and brass
  General machine work
  Gas engines
  Locomotives
  Machine tools
  Molding machines
  Pumps
  Pneumatic tools
  Sewing machines
  Typewriters
  Wood-working machinery
Metal and coal mining
Metal working
  Bolts and nuts
  Chains
  Hardware
  Tanks
  Tin cans
  Valves and pipe fittings
Miscellaneous manufacturing
  Beer
  Beet sugar
  Boxes (wood and paper)
  Buttons
  Clothing
  Cordage
  Food products
  Furniture
  Flour
  Glass
  Lumber products
  Pianos
  Paper and paper pulp
  Rubber goods
  Soaps

Shoes
Slate products
Printing and lithographing
Railroad maintenance of motive power
Steel manufacture
Textile manufacture
Bleaching and dyeing
Cottons
Velvets
Woolens

MINORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION.

I am unable to sign the majority report in its entirety, much as I admire the thoroughness with which it has been prepared and its great interest. In its general tenor it distinctly implies the desirability of what is termed labor-saving management, involving the planning department, functional organization and the bonus system. Perhaps this statement is not strictly justified, but I cannot avoid the impression after reading the report most carefully. That the methods of management have undergone a great change in recent years I certainly agree. I would explain it by stating that in many respects the art of management is developing into a science. In common with most lines of work, the method of investigating facts has changed. Phenomena are analyzed, information is obtained accurately, the mental attitude of the manager is scientific rather than empirical. Things that used to be known generally by gradual experience are now known specifically by detailed observation. In the course of this development many new systems and ideas have been invented, time studies, motion study, payment systems, functional management, etc. Some are new, others partly new, others simply practices of many managers put into definite form. The science of management will classify these for us, perhaps even explain their advantages and limitations so that we may in time know which is preferable and when.

2. But the introduction of the use of these methods has been attended by claims as to the results that might be obtained by their use, and these claims have led to such absurd statements as that made before the Interstate Commerce Commission, which is referred to in the report. I feel most strongly that each of the suggestions that have been made to improve methods of management may have merit, but I do not feel that any one of them is a panacea for all our inefficiency. For instance, in certain classes of work time studies are valuable, in others they may be a waste of time. In certain classes of work I consider piece-work or bonus systems desirable, in others I consider them inferior to day work. Functional management may be an improvement in certain industries, in others I do not consider it suitable. In some cases the workman can be trained by the skilled executive, in others he can train him with ease.

3. In general I feel that labor-saving management is not any particular system, but will always remain the art of selecting and applying the most appropriate methods furnished by the science of management, the science that records what these methods are and the results obtained from them. As you will see, I agree with your correspondent in his last paragraph, quoted under Par. 28; but his opinion is not really represented in the trend of the report.

H. H. VAUGHAN,
*Member, Sub-Committee on Administration.*

## NEW PUBLICATIONS OF THE BUREAU OF MINES OF THE DEPARTMENT OF THE INTERIOR.

### BULLETINS.

Bulletin 43. Comparative fuel values of gasoline and denatured alcohol in internal-combustion engines, by R. M. Strong and Lauson Stone. 1912. 243 pp. 3 pls.

Bulletin 46. An investigation of explosion-proof motors, by H. H. Clark. 1912. 44 pp. 6 pls.

### TECHNICAL PAPERS.

Technical Paper 25. Methods for the determination of water in petroleum and its products, by I. C. Allen and W. A. Jacobs. 1912. 13 pp.

Technical Paper 28. Ignition of gas by standard incandescent lamps, by H. H. Clark. 1912. 4 pp.

The Bureau of Mines has copies of these publications for free distribution, but can not give more than one copy of the same bulletin to one person. Requests for all papers can not be granted without satisfactory reason. In asking for publications please order them by number and title. Applications should be addressed to the Director of the Bureau of Mines, Washington, D. C.

# SHIPS.

## AUSTRIA.

The Austrian *Viribus Unitis* has exceeded her designed speed for about three-quarters of a knot. She was designed for 21 knots and on trial made 21.8. This may or may not dispose of the rumor that she floats a good deal deeper than intended, as some ships steam better in such circumstances. This was very noticeable in the trials of the German *Nassaus*, which are known to draw a good deal more than the designed draught. It is officially stated in Austria that the *Viribus Unitis* merely draws one and a-half inches over the designed amount.—"The Engineer."

## FRANCE.

### TRIALS OF THE FRENCH DESTROYERS *FOURCHE* AND *FAUX*.

The French destroyers *Fourche* and *Faux* are noteworthy on account of the fact that both vessels are of purely French design as regards hull and propelling machinery. Their dimensions are:

| | |
|---|---|
| Length overall, feet and inches | 247–11 |
| Breadth, feet and inches | 25–01 |
| Depth, feet and inches | 15–11 |
| Draught fully loaded, feet and inches | 9–07 |

The form of stern is peculiar and resembles that of the Laubeuf type of submarine. The propelling machinery consists of Rateau-Chantiers de Bretagne turbines driving twin screws. Steam is supplied by four Du Temple water-tube boilers. The contract speed for the six hours' full-speed trial was 31 knots. The particulars of the official trials are given in Tables II and III.

TABLE II.—SIX-HOURS' FULL-SPEED TRIALS.

| | *Fourche.* | *Faux.* |
|---|---|---|
| Displacement, tons | 725 | 722 |
| Draught, feet and inches | 8–03 | 8–03 |
| Pressure at boilers per square inch, pounds | 228 | 228 |
| Pressure at steam chest per square inch, pounds | 171 | 171 |
| Revolutions per minute | 680 | 658 |
| Mean speed, knots | 33.20 | 32.01 |
| Speed on best run, knots | 33.60 | 34.90 |
| Shaft horsepower | 18,500 | 18,500 |
| Consumption of fuel per hour, tons | 10.25 | 10.1 |
| State of sea | Smooth | Rough |

TABLE III.—EIGHT-HOURS' FUEL-CONSUMPTION TRIAL.

| | *Fourche.* | *Faux.* |
|---|---|---|
| Displacement, tons | 725 | 722 |
| Draught, feet and inches | 8–03 | 8–03 |
| Number of boilers in use | Two. | Two. |
| Pressure at boilers per square inch, pounds | 228 | 228 |
| Pressure at steam chest per square inch, pounds | 114 | 114 |
| Revolutions per minute | 242 | 242 |
| Mean speed, knots | 14 | 14 |
| Consumption of fuel per hour, tons | 0.90 | 0.91 |

On her preliminary trials the *Faux* attained a mean speed of 33.37 knots, and would undoubtedly have done better than she did on the official trials had the weather been finer. Her bottom was also foul owing to the fact that she had not been docked and painted within two months previous to the trials.—"The Shipbuilder."

## GERMANY.

The new German protected cruiser *Magdeburg* made 27.5 knots on trials. The designed speed was 25.5. A feature of these cruisers, as opposed to their predecessors, is that they are a good deal longer and have four funnels instead of three. The armament, protection and coal capacity remain pretty much as they were before. It is curious that the Germans in ships of this sort should still rely on the 4.1-inch gun in complete disregard of the fact that our corresponding ships now carry nothing but 6-inch.—"The Engineer."

## GREAT BRITAIN.

### H.M. TORPEDO-BOAT DESTROYER *LURCHER*.

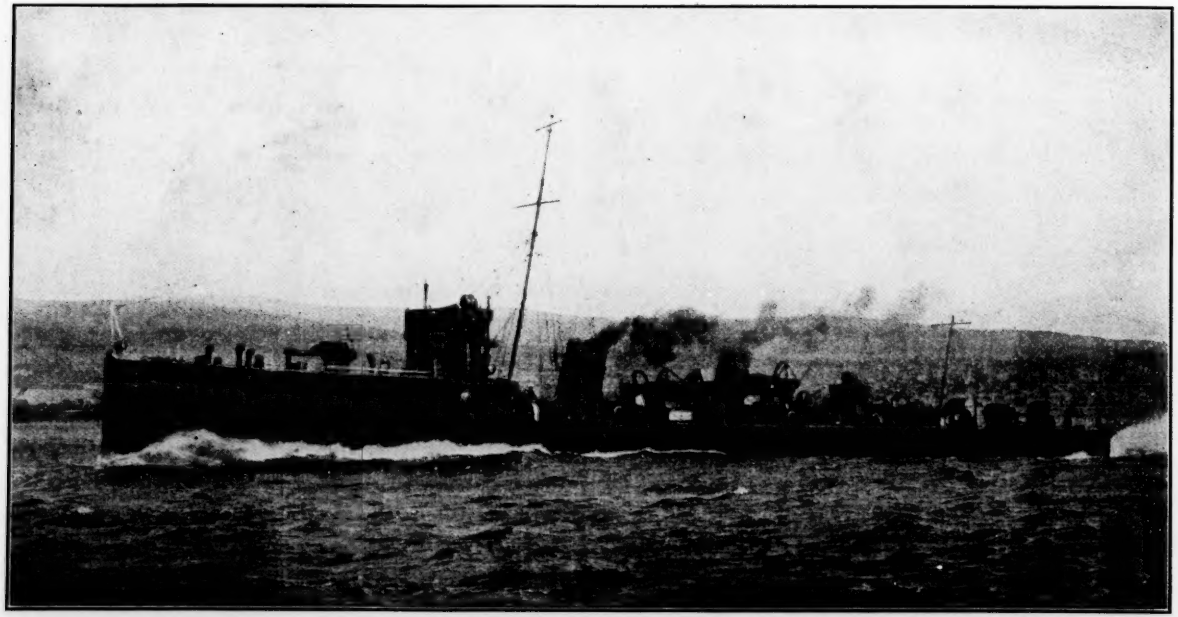
H.M. torpedo-boat destroyer *Lurcher* has just attained remarkably successful results on her speed trials. She is one of three ships of a special type ordered at the beginning of last year from Messrs. Yarrow & Co., of Glasgow. The *Lurcher*, which forms one unit of the *Firedrake* type, obtained during an official full-speed trial of eight hours a mean speed of 35.345 knots, or 3.345 knots in excess of the contract speed of 32 knots. It should be mentioned that the trial was run in deep water.

Much has been heard of late as to the results obtained with Continental torpedo-boat destroyers, and on certain occasions a comparison somewhat detrimental to British construction has been made. The above result, however, removes grounds for such disparagement; no destroyer has ever attained such a high speed during a continuous run of eight hours. Thus the performance proves and accentuates the superiority of British destroyers.

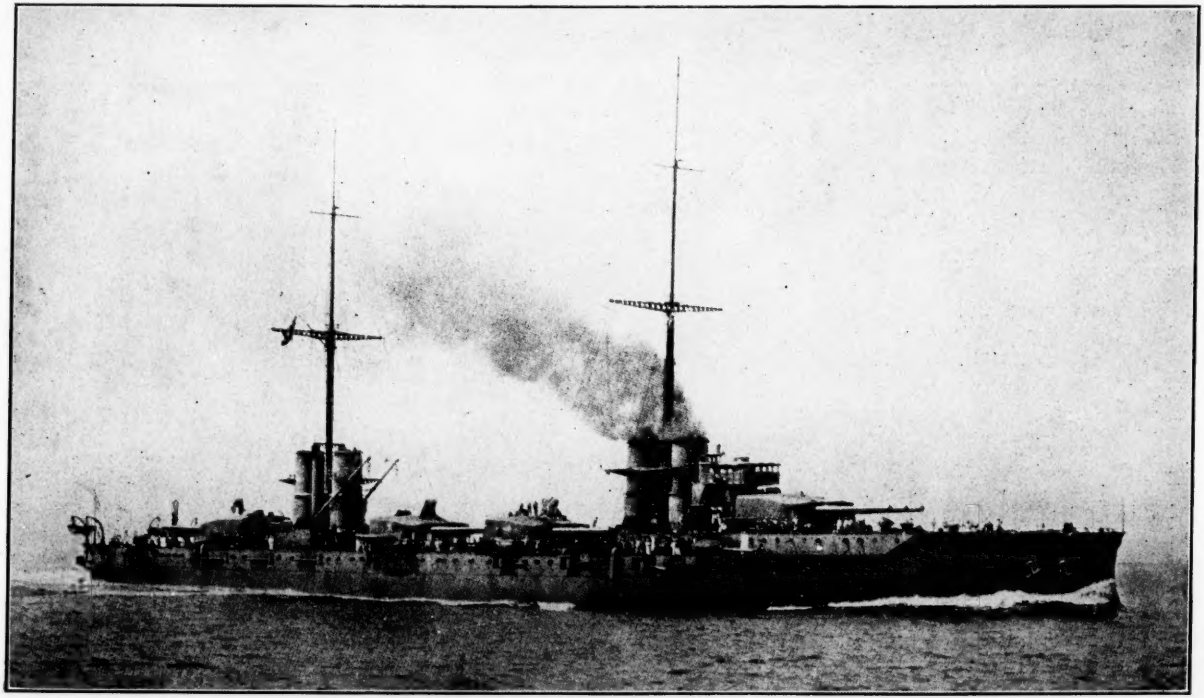
The *Firedrake* type of destroyer is 255 feet long by a beam of 25 feet 7 inches. The propelling machinery consists of Parsons turbines driving twin screws, steam being supplied by three of the latest type of Yarrow boilers, fitted with the firm's patent feed-heating device, and arranged for burning oil fuel only. It may be stated that the *Firedrake* type represents an important advance in destroyer construction, and clearly justifies the action of Admiral Sir John Jellicoe, who, as Controller of the Navy, placed the order for these three special vessels with a firm which specializes in the construction of this type of craft.—"Engineering."

### H.M.S. *PRINCESS ROYAL*.

Those people who rejoice in the holding of records will have heard with pleasure of the success of our latest battle cruiser the *Princess Royal*. Two weeks ago she went out for her full-power trials off Plymouth, and attained the high speed for a vessel of her type of 32 knots. But that was not the best she could do. On her return she was dry-docked at Devonport and her propellers were changed. She coaled and went out again for six runs at three-fourths and six at full power on the Polperro mile. On this occasion it is reported that she reached a speed of no less than 34.7 knots and made an average of 33, which establishes a new world's record for vessels of her class. We understand that three-



H.M. Torpedo-Boat Destroyer "Lurcher."



ITALIAN DREADNOUGHT "DANTE ALIGHIERI."

Rivista Marittima.

bladed propellers were used in the second trials. The *Princess Royal* is a sister to the *Lion*. She has a maximum displacement of 26,500 tons, is 700 feet long, has a beam of 88 feet 6 inches, and her turbines develop an aggregate horsepower of 70,000. She was launched at Barrow on April 29th, 1911, and was designed for an official speed of 28 knots, but with full anticipation that she would do more. Her boilers are Yarrow type. She carried eight 13.5 guns in pairs and twenty 4-inch guns. It is interesting to note that the original maximum speed of the *Lion* was 29 knots, but that she has subsequently made 31.7 knots.—"The Engineer."

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## ITALY.

### TRIALS OF THE *DANTE ALIGHIERI*.

Translated from "Revista Marittima," by A. CONTI, Associate.

The *Dante Alighieri* is a first-class battleship of the dreadnought type, of about 19,000 tons normal displacement, 520 feet long and 87.5 feet beam.

Her main battery consists of twelve 12-inch guns, arranged in four turrets on the central line of the ship, and her intermediate battery comprises twenty 4-inch guns, of which eight are mounted in pairs in four turrets located on the main deck. The main propelling machinery was supplied by the Ansaldo-Armstrong Co., of Genoa, and consists of Parsons' turbines on four shafts, arranged in three engine rooms. The boiler plant consists of twenty-three water-tube boilers of the Blechynden type.

Her first preliminary trial took place on June 21st; it lasted three hours, and with only six boilers in operation, a mean speed of 11.46 knots was obtained with 3,408 S.H.P.

A 12-hour official trial to ascertain the maximum radius of action, using only six boilers, was held June 24, and gave a mean speed of 11.29 knots with about 3,000 S.H.P. The hourly fuel consumption averaged 2.31 pounds of coal per S.H.P., which includes a certain amount of liquid fuel expressed in its equivalent of coal.

A preliminary full-power trial was run on June 27th with the following results: At natural draft: speed 22.15 knots; S.H.P., 25,400. At forced draft: speed 23.58 knots; S.H.P., 34,200.

The official six-hour full-power trial bettered these results, giving a mean speed of 22.83 knots, with 31,460 S.H.P. The equivalent fuel consumption was 1.475 pounds of coal per S.H.P. hour.

On July 16 and 17 a twenty-four-hour endurance trial took place. The first eighteen hours were run at natural draft, at a mean speed of 19.8 knots and 20,220 S.H.P. The equivalent fuel consumption averaged 1.545 pounds of coal per S.H.P. hour. The last six hours of this trial were run at varying speeds and included a double run over a measured course during which a speed of 23.825 knots was obtained with 34,860 S.H.P.

We recall that the *Dante Alighieri* was built at the Castellamare Navy Yard and was completed at the Spezia Navy Yard. The keel was laid June 6th, 1909; she was launched August 20th, 1910, and has thus been built and completed her trials within 37 months from the laying of the keel, a fact which marks a remarkable progress in the rapidity of construction for the Italian Navy.

### TRIALS OF THE SCOUT CRUISER *QUARTO*.

The *Quarto* is a new scout cruiser of 3,230 tons normal displacement. She is 413 feet long between perpendiculars and has a beam of 42 feet. Her designed speed is 28 knots. Her propelling machinery consists of

Parsons' turbines on four shafts and Blechynden water-tube boilers, burning fuel oil exclusively.

A ten-hour official trial to determine her cruising radius was held August 2d; the mean speed obtained was 15.2 knots with about 3,850 S.H.P. Her fuel consumption averaged 1.74 pounds of crude oil per S.H.P. hour.

The next official trial was run August 6 and 7, and consisted of a twenty-four-hour run at half power. An average speed of 22.29 knots was obtained with about 12,500 S.H.P. and a liquid-fuel consumption of 1.20 pounds per S.H.P. hour.

A four-hour full-power official gave the following average results: Speed, 28.7 knots; R.P.M., 577; S.H.P., 28,600; fuel-oil consumption, 1.10 pounds per S.H.P. hour. Credit for the fine performance of this ship is due to her designer, Major Truccone, formerly of the Corps of Naval Constructors, who also supervised her construction at the Royal Navy Yard of Venice.

*Note by Translator.*—The results of the *Quarto's* trials will furnish an interesting comparison when the performances, of the two scouts, *Nino Bixio* and *Marsala,* of similar type, are made public, inasmuch as these two ships are fitted with steam turbines of the Curtis type, on three shafts, aggregating 22,500 S.H.P., which were designed and built in this country. The trials of the first of these two vessels will take place December next.

## TRIALS OF THE TORPEDO-BOAT-DESTROYER *INDOMITO.*

The trials of the *Indomito* are of particular interest, as she is the first vessel of a class of six units, equipped with the marine type of Tosi turbines. These ships are 240 feet long and have a beam of 24 feet.

A series of progressive trials were run over a measured course from a minimum speed of 13.35 knots, 217 R.P.M. and 600 S.H.P., to a maximum speed of 34.56 knots, 678 R.P.M. and 16,580 S.H.P.

In a three-hour full-power trial a mean speed of 35.06 knots was obtained with 686 R.P.M. and 16,660 S.H.P. The fuel consumption averaged 1.31 pounds of liquid fuel per S.H.P. hour.

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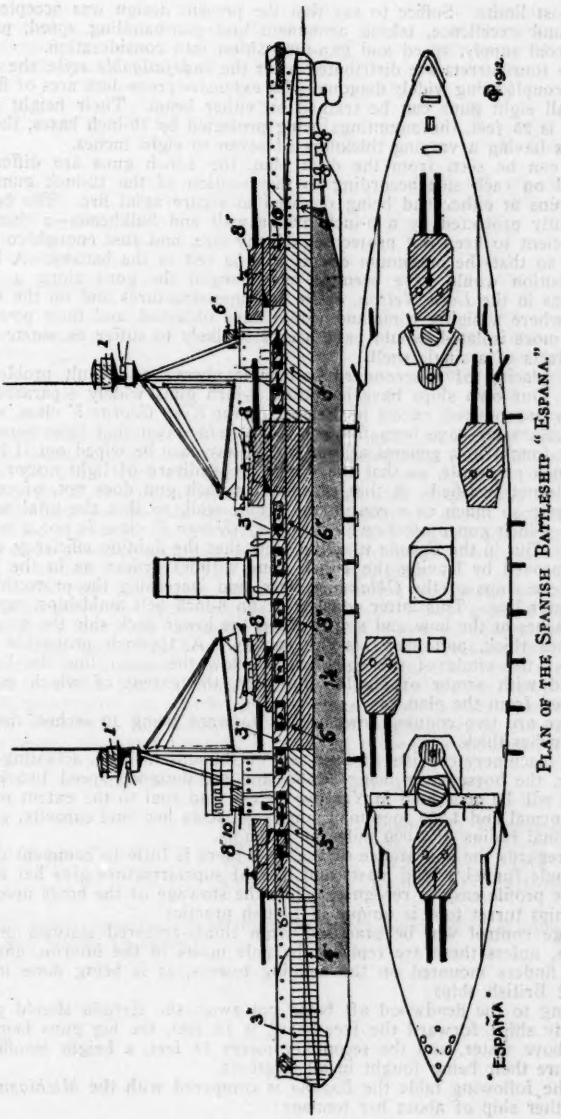
## SPAIN.

The three battleships of the *España* class, the *España, Alfonso XIII* and *Jaime I,* now building in Spain, under the direction of the "Vickers-Elswick" syndicate, represent a type which might be classed as second-class *Dreadnoughts.* The contract for their construction was secured in open competition for the best design within given limitations, the times of building to be four years, five-and-a-half years, and seven years respectively from the date of contract. During the first two years, 75 per cent. of the workmen engaged were to be "foreign," and from then onwards only 50 per cent., the remainder being Spanish.

The type represents an interesting attempt to produce an all-big-gun ship upon moderate dimensions, the displacement being 15,700 tons, length (water line) 439½ feet, beam 78¾ feet, and mean draught 26 feet. The armament consists of eight 12-inch 50-caliber guns in four twin turrets, and twenty 4-inch guns mounted in a main-deck battery, together with four smaller pieces and three submerged torpedo tubes. Particulars of the competitive designs sent in have not been made public, otherwise it would have been interesting to have seen what the different European firms could have produced upon the displacement




PLAN OF THE SPANISH BATTLESHIP "ESPANA."

and cost limits. Suffice to say that the present design was accepted for all-round excellence, taking armament and gun-handling speed, protection, coal supply, speed and gun-disposition into consideration.

The four turrets are distributed after the *Indefatigable* style, the amidships couple being widely diagonal with extensive cross-deck arcs of fire, so that all eight guns can be trained on either beam. Their height above water is 25 feet, the mountings being protected by 10-inch bases, the gun houses having a varying thickness of seven to eight inches.

As can be seen from the deck plan, the 4-inch guns are differently spaced on each side according to the position of the 12-inch guns, the two guns at either end being recessed to secure axial fire. The battery is lightly protected by a 3-inch armor wall and bulkheads—a thickness insufficient to keep out projectiles of any size, and just enough to burst shells so that the maximum effects will be felt in the battery. A better distribution would have been to have ranged the guns along a flying deck, as in the *Lord Nelson*, or in the superstructures and on the turret tops, where a higher command would have obtained, and their positions, being more isolated, would have been less likely to suffer *en masse* from the effects of a single shell.

The placing of a secondary battery is always a difficult problem to solve. Our own ships have had their 4-inch guns widely separated and usually unprotected, except in the case of the *King George V* class, where light casemates have been fitted, it being laid down that light guns will not be fought in a general action, and in any case be wiped out if hit by a 12-inch projectile, so that the weight expenditure of light armor upon them is not justified. A thin casemate to each gun does not, of course, aggregate so much as a continuous armor wall, so that the total weight of the 4-inch gun protection in the *King George V* class is not a serious factor. But in the *España* it would seem that the fighting efficiency would be improved by leaving the 4-inch guns without armor, as in the latest American ships of the *Oklahoma* class, and increasing the protection on the water line. This latter consists of an 8-inch belt amidships, tapering to 4 inches at the bow and stern; along the lower deck side the armor is 6 inches thick, and above is the battery. A 1½-inch protective deck encloses the vitals of the ship, while below the water line the hull is covered with armor of similar thickness, the extent of which can be gathered from the plan.

There are two conning towers, that forward being 10 inches, the one aft 6 inches thick.

The machinery consists of four sets of Parsons turbines actuating four screws, the horsepower being 15,500, and the designed speed 19.5 knots. Steam will be generated in Yarrow boilers, and coal to the extent of 900 tons normal and 1,900 tons maximum represents her fuel capacity, giving a nominal radius of 5,000 miles at 10 knots.

As regards the appearance of the ship, there is little to comment upon; the single funnel, tripod masts and lack of superstructure give her a distinctive profile easy to recognize, while the stowage of the boats upon the amidships turret tops is unique in British practice.

Range control will be practised from thinly-armored stations on the tripods, unless these are replaced by pole masts in the interim, and the range finders mounted on the conning towers, as is being done in the newest British ships.

Owing to the deadwood aft being cut away, the *España* should prove a handy ship; forward the free-board is 18 feet, the big guns being 25 feet above water, and the secondary pieces 14 feet, a height insufficient to assure their being fought in all weathers.

In the following table the *España* is compared with the *Michigan*, the only other ship of about her tonnage:

| | *España* | *Michigan.* |
|---|---|---|
| Displacement | 15,700 tons | 16,000 tons |
| Dimensions | 439×78¾×26′ | 450×80¼×24½′ |
| Complement | 700 men | 869 men |
| Armament | 8 12″<br>20 4″<br>8 small<br>3 T. Tubes | 8 12″<br>22 3″<br>2 smaller<br>2 T. Tubes |
| Protection: | | |
| Belt | 8″—4″ | Belt, 11″—9″—1½″ |
| Lower deck | 6″ | 10″—8″ |
| Deck | 1½″ | 3″ |
| Main deck | 3″ | Nil |
| Under water | 1½″ | Nil |
| Conning tower | (a) 10″<br>(b) 6″ | (a) 12″ |
| I.H.P. | 15,500 | 16,500 |
| Designed speed | 19.5 knots | 18.5 knots |
| Machinery | Turbine | Reciprocating |
| Coal | 900/1900 tons | 900/2,380 tons |
| Launched | 1912 | 1908 |

The two ships represent the "all center-line" and "*en echelon*" theories for eight guns, and offer a wide field for comparison. With all her big guns on the same level, the *España* possesses a more uniform command than does the U. S. ship, which was fully dealt with in our issue of July, 1908. She can also fire six guns ahead or astern, on paper, against the four of the *Michigan*, but on the beam the latter has the advantage, as all her eight guns have uninterrupted arcs of training in contrast to the limited cross-deck bearing of two of the *España's* guns.

The U. S. ship has better water-line protection and more armor on her 12-inch guns, an open secondary battery of 3-inch guns, a larger coal supply, a knot less designed speed, which, however, counts for little, as twenty knots has been reached on trial, and presents a bigger target. She has, however, "soft" ends, the belt being 1½ inches aft and forward to within some 22 feet of the bow, leaving the cut-water open to shell destruction, in which case her speed and stability would be seriously impaired. The secondary battery of 14-pounders is distributed well up above the water line, but is too light to destroy modern torpedo craft of any size.

The dates of completion of the three ships are as follows:

| | Laid down. | Launched. | Completed. |
|---|---|---|---|
| *España* | Dec., 1909 | Feb., 1912 | 1913 |
| *Alfonso XIII.* | Feb., 1910 | Building | 1914 |
| *Jaime I.* | Feb., 1912 | Building | 1915–16 |

All are under construction at the new Ferrol dockyard.—"Marine Engineer and Naval Architect."

# ASSOCIATION NOTES.

The Society was officially represented at the Sixth Congress of the International Association for Testing Materials, held at New York City during the first week in September, 1912. Its delegate was Captain Clarence A. Carr, U. S. Navy.

At a meeting of the Council of the Society held on Friday, September 13, 1912, Lieutenant Commander John Halligan, Jr., U. S. Navy, tendered his resignation as Secretary-Treasurer of the Society on account of his detachment from duty in Washington. The resignation was accepted to take effect on September 15, 1912. Lieutenant Commander E. L. Bennett, U. S. Navy, was elected to serve the uncompleted term as Secretary-Treasurer until December 31, 1912, inclusive.

On September 27, 1912, at a meeting of the Council, Captain Emil Theiss, U. S. Navy, was unanimously elected as a member of the Council of the Society to serve until the next regular election of officers. Captain Theiss was elected to fill the vacancy in the Council caused by the resignation of Lieutenant Commander John Halligan, Jr., U. S. Navy.

In accordance with its By-Laws, a regular meeting of the Society was held on Tuesday, October 1, 1912, for the purpose of nominating officers for the year 1913, and for the transaction of such other business as might come before the meeting. The following members were nominated for officers for the year 1913:

For President:
Captain R. S. Griffin, U. S. N.
For Secretary-Treasurer:
Lieutenant O. L. Cox, U. S. N.

For Members of Council (three to be voted for):

Read Admiral John R. Edwards, U. S. N.
Engineer-in-Chief C. A. McAllister, U. S. R. C. S.
Captain Gustav Kaemmerling, U. S. N.
Captain Emil Theiss, U. S. N.
Lieutenant Commander Chester Wells, U. S. N.
Lieutenant Commander E. L. Bennett, U. S. N.
Lieutenant Commander D. W. Todd, U. S. N.
Lieutenant Commander D. F. Boyd, U. S. N.
Lieutenant George B. Wright, U. S. N.

The committee appointed to audit the accounts of Lieutenant Commander John Halligan, Jr., U. S. N., as Secretary-Treasurer of the Society, reported that the audit had been made and the books found correct.

It was voted to hold a banquet during the early part of 1913. The following members were appointed a committee with power to select the date and to make all necessary arrangements for the banquet:

Captain William Strother Smith, U. S. N.
Engineer-in-Chief C. A. McAllister, U. S. R. C. S.
Lieutenant N. H. Wright, U. S. N.
Lieutenant O. L. Cox, U. S. N.

The following new members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Blakely, Charles A., Lieutenant, U. S. N.
Brainard, Roland M., Lieutenant, U. S. N.
Davis, Guy E., Lieutenant, U. S. N.
Delano, Harvey, Lieutenant, U. S. N.
Hammond, Philip H., Lieutenant, U. S. N.
Parsons, Archibald L., Civil Engineer, U. S. N.
Stoddard, George K., Ensign, U. S. N.

ASSOCIATES.

Behr, F. J., Captain, U. S. Coast Artillery, Fort Monroe, Va.
Brew, William P., Westinghouse Machine Co., 165 Broadway, New York City.
Emerson, Joshua, Jr., 913 Sutter St., Vallejo, Cal.
Hanker, F. C., 500 South Avenue, Wilkinsburg, Pa.
Herr, H. T., Second Vice President, Westinghouse Machine Company, East Pittsburgh, Pa.
Nichols, John F., Assistant Chief Engineer, Newport News Shipbuilding and Dry Dock Co., Newport News, Va.

The annual meeting of the Society for the counting of votes for officers for 1913 will be held in the Bureau of Steam Engineering on Monday, December 30, 1912, at 4:30 P. M.